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DEVELOPMENT OF AN ANTI-JACKKNIFING SYSTEM FOR TRACTOR SEMITRAILERS : FEASIBILITY STUDY, PHASE 1

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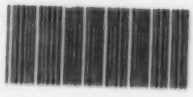

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DEVELOPMENT OF AN ANTI-JACKKNIFING SYSTEM FOR TRACTOR SEMITRAILERS: FEASIBILITY STUDY, PHASE 1

By

Suanne L. Dorion, Henry E. Vietinghoff, and James G. Pickard

T E S Limited

June 1989

DDN 434529

The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transportation Development Centre of Transport Canada.

Un sommaire en français de ce rapport est inclus avant la table des matierès.

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16. Abstract

This study investigated the feasibility of developing an anti-jackknifing system, located about the fifth wheel, to control jackknifing and trailer swing of tractor semitrailers. Initially, information was gathered on the jackknifing phenomenon, jackknifing accidents and anti-jackknifing systems (AJS). From this a generic performance classification of AJS technology was developed.

The characteristics of jackknifing and the performance of the various AJS were determined by computer simulation. Several different vehicle configurations representative of the wide range of tractor semitrailers on the road today were used in the analysis. Steady-state handling characteristics, in the form of handling diagrams and articulation angle response curves, were used to aid in the analysis of these vehicles. Performance evaluation of the AJS devices was based mainly on the change in vehicle articulation angle within a 1.5 second driver reaction time, the intrusion of the vehicle into a neighbouring lane, the lateral stability of the vehicle, and the resultant path of the vehicle.

It was concluded that several types of AJS showed merit in controlling jackknifing but had a minimal or negative impact on trailer swing.

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Feasibility of Anti-Jackknifing Systems for Tractor Semitrailers, SAE 891631, August, 1989.

16. Résumé

Cette recherche avait pour but d'étudier la faisabilité de divers systèmes conçus pour prévenir les mises en charnière des véhicules articulés et les oscillations des semi-remorques en agissant sur la sellette. La première étape a été de lancer une recherche documentaire dans le but de cerner le phénomène des mouvements latéraux induits et de recueillir de l'information sur les accidents qui leur sont attribuables et sur les dispositifs antimises en charnière existants. Elle a débouché sur une classification générale des dispositifs en fonction de leurs performances.

La détermination des conditions propices aux mises en charnière et l'évaluation des performances relatives des divers systèmes examinés ont été faites par simulation assistée par ordinateur. Plusieurs configurations formées par différents types de véhicules articulés, et représentatives de la vaste gamme de combinaisons possibles, ont été prises en compte. L'analyse de la pilotabilité des véhicules articulés a nécessité la réalisation de graphiques représentant l'évolution des caractéristiques de stabilité et des angles d'articulation dans le temps. Les principaux critères d'appréciation des systèmes antimises en charnière ont été leur efficacité de détecter les changements d'angle d'articulation à l'intérieur d'une plage de 1,8 seconde, valeur représentant le temps de réaction type du conducteur, le départ intempestif sur une voie parallèle, la stabilité dynamique transversale du véhicule et leur effet sur la trajectoire suivie par celui-ci.

La recherche a montré que certains dispositifs pouvaient prévenir les mises en charnière dans une certaine mesure mais que leur efficacité pour contrer les oscillations des semi-remorques a été faible s'ils n'ont pas été carrément nuisibles.

17. Mots clés	18. Diffusion
Systèmes antimise en charnière, dispositifs antimise en charnière, mise en charnière, oscillations des semi-remorques, simulation assistée par ordinateur, véhicules articulés.	Le Centre de développement des transports dispose d'un nombre limité d'exemplaires.

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SUMMARY

In October 1988, T E S Limited was contracted by the Transportation Development Centre to conduct a feasibility study of anti-jackknifing systems (AJS) for tractor semitrailers. This study was initiated as a result of the serious safety problems associated with jackknifing which every year contributes to a significant number of fatalities, personal injuries and property damage in Canada. Jackknife accidents make up 6% to 9% of all combination vehicle accidents, of which the latter represent approximately 2% of all vehicle accidents in Canada and the U.S.

The first step of the project was to gather information related to the jackknifing phenomenon. Literature searches, covering both national and international developments, provided valuable information on anti-jackknife devices, vehicle dynamics and accident statistics. A Canadian and U.S. patent search revealed that hundreds of patents related to anti-jackknife devices have been awarded. A sample cross section of approximately 30 patents was obtained and studied in detail. Comments, opinions, and suggestions were collected from numerous people in the trucking industry, including truck drivers, fleet owners, manufacturers, government and its related institutions.

The study included analysis of vehicle dynamics and the results of computer simulations of vehicles manoeuvring a constant radius turn and a lane change. Two tractors and two semitrailers were chosen for study such that, when combined, they would produce four different combinations representative of a wide range of tractor semitrailers on the road today. The primary tool used in the study was a computer simulation program, the "HSRI/MTC Simulation Program for a Tractor-Trailer in the Horizontal Plane", which was chosen from a list of eleven that were examined. Computer simulation provided a means of analysis in three areas:

1) Steady-state handling characteristics,
2) determination of the characteristics of jackknife and trailer swing, and,
3) evaluation of jackknife control methods.

Determination of the steady state handling characteristics involved constructing articulation angle response curves and handling diagrams for the configurations. Over 130 simulations were required to complete the diagrams, which provided insight into the relative stabilities or instabilities of the vehicles.

Computer simulation of the configurations completing the constant radius turn and the lane change manoeuvres in the absence of braking, without jackknife or trailer swing and with no anti-jackknife device installed, provided "ideal" performance against which other simulation runs could be compared. "Worst case" performance was developed by inducing jackknife and trailer swing through lock up of the rear tractor tires and semitrailer tires respectiyvely. These initial simulations provided a performance range against which to compare subsequent simulations incorporating the use of the anti-jackknife devices. The simulation runs with induced jackknife and

trailer swing also provided data which aided in the determination of the specific characteristics of these two modes of instability.

The investigation showed that tractor jackknife and trailer swing are instabilities resulting from saturation or limitation of the lateral forces at the tire road interface. Trailer swing proved to be a milder form of instability as it tends to regain stability without any form of compensation. Factors that influence the severity of jackknife or trailer swing are mainly configuration dependent. An increase in the inertia of either vehicle unit, tractor or semitrailer, will slow the jackknife or trailer swing rate because of inertial resistance, but will produce a larger torque about the fifth wheel. Conversely, lower inertia is associated with a more rapid development of jackknife or trailer swing, but produced lower torque values about the fifth wheel. Any increase in velocity will increase the severity of jackknife or trailer swing.

Free body analysis gave an approximate range of torques about the fifth wheel that arise during jackknife and trailer swing. It was calculated that a long wheelbase tractor with an empty 14.7 metre (48 feet) long semitrailer produced torques about the fifth wheel of the order of 95,000 N•m in jackknife and 140,000 N•m in trailer swing. Increases in payload had a more significant effect on the torques produced in trailer swing than those produced in jackknife.

Sensing the onset of jackknife and trailer swing proved to be difficult because the characteristics of each are often duplicated in the transient responses of the vehicle during normal turning manoeuvres. Fifteen degrees of articulation is a commonly accepted value used to indicate the occurrence of jackknife and trailer swing during highway travel. This was, however, found to be a late indicator. Other parameters such as the tractor or semitrailer yaw rates are earlier indicators, but are difficult to measure in practice. The relationship between certain characteristics or parameters of the vehicle may be used to indicate the onset of jackknife or trailer swing, such as articulation rate and vehicle speed. Unfortunately these are configuration dependent and do not readily lend themselves to tractor semitrailer applications where vehicle units are frequently interchanged.

Potential methods of controlling jackknife and trailer swing were developed from the information gathered from the background research. The impact of these various methods on vehicle stability were determined through computer simulation. The control methods were characterized as locking, damping, limiting or a combination thereof. Locking prevents any further articulation at the onset of jackknife or trailer swing. Damping, which can take many forms, slows down the articulation thereby providing the driver with more time to react. Limiting restricts the articulation to a specified range.

Four devices were modelled; constant torque device (approximation of locking), Safe-T-King (limiting), Breeze (form of damping), and a variable dampening device. All four devices showed merit in controlling jackknife. The relative performance of the devices, in an initial 1.5 second time window, may be rated in the following order:

1. Constant Torque device,
2. Safe-T-King device,
3. Variable damping device, and
4. Breeze device.

The 1.5 second time window is used as this represents a conservative driver reaction time that is typical of night time driving.

The devices studied showed little or no merit in controlling trailer swing. Most presented a worse situation than if no device had been applied. This adverse impact is due to the large torques produced in trailer swing that saturate the tractor rear tires when articulation is prevented. The result is a tractor that rotates out of control in conjunction with the trailer.

The study results lead to the recommendation that future research be concentrated on attempting to distinguish jackknife from trailer swing and controlling jackknife only. This is in recognition of the following:

1) when devices are used to control trailer swing an adverse situation results;

2) trailer swing tends to regain stability on its own; and

3) jackknife is attributed to a larger portion of fatal accidents than trailer swing.

SOMMAIRE

En octobre 1988, le Centre de développement des transports de Transports Canada a donné à T E S Limited le mandat de mener une étude de faisabilité sur divers systèmes pour la prévention des mises en charnière des véhicules articulés. Cette étude a été motivée par les graves accidents reliés à ce phénomène et qui sont à l'origine chaque année, de nombreux tués, ainsi que des dommages corporels et matériels importants au Canada. Les accidents attribuables aux mises en charnière comptent pour 6 à 9 p. 100 de tous les accidents de véhicules articulés, lesquels représentent quelque 2 p. 100 de tous les accidents routiers au Canada et aux Etats-Unis.

La première étape a été de lancer une recherche documentaire sur ce phénomène, tant à l'échelle nationale qu'à l'échelle internationale. Cette étude a permis de recueillir une abondante information sur les dispositifs de prévention des mises en charnière, la stabilité dynamique des véhicules articulés et l'accidentologie afférente. Une recherche sur les brevets au Canada et aux Etats-Unis a montré que plusieurs centaines de brevets ont été délivrés dans ce domaine précis. Un échantillonnage de quelque 30 brevets a été obtenu et étudié par les chercheurs qui, par ailleurs, ont recueilli les commentaires, les opinions et les suggestions des divers milieux concernés : conducteurs et propriétaires de camions, fabricants, pouvoirs publics et services gouvernementaux.

La recherche s'est intéressée au domaine de la stabilité dynamique des véhicules articulés effectuant des changements de file ou des virages à rayon de braquage constant et aux résultats des simulations assistées par ordinateur. Deux tracteurs et deux semi-remorques ont été mobilisés pour la recherche, permettant d'obtenir quatre combinaisons possibles qui sont représentatives des combinaisons que l'on voit circuler le plus souvent. Outil principal de cette recherche, le programme de simulation assistée par ordinateur "HSRI/MTC Simulation Program for a Tractor-Trailer in the Horizontal Plane" avait été sélectionné parmi la dizaine de programmes intéressants. Ce programme permet :

1) d'analyser la pilotabilité des véhicules articulés en régime permanent,
2) de déterminer le mécanisme des mises en charnière et des oscillations des semi-remorques,
3) d'évaluer les diverses méthodes de lutte contre ces phénomènes.

L'analyse de la pilotabilité en régime permanent a nécessité la réalisation de graphiques représentant l'évolution des caractéristiques de stabilité dynamique et des angles d'articulation dans le temps, et ce pour chacune des configurations prises en compte. Les quelque 130 simulations nécessaires ont permis d'approfondir les conditions de stabilité ou d'instabilité dynamique des véhicules.

Les simulations faites ont d'abord représenté des configurations effectuant des changements de file et des virages à rayon de braquage constant, sans freinage, sans mise en charnière, sans oscillations de la semi-remorque et sans dispositif antimise en charnière. Ainsi, des courbes "idéales" ont pu être obtenues afin de servir de base de comparaison pour les simulations ultérieures. Les cas de simulation les plus "défavorables" ont représenté des configurations soumises à des mouvements latéraux - mise en charnière et oscillations de la semi-remorque - induits par le blocage des roues des essieux moteurs et des essieux porteurs, respectivement. Ces simulations préalables ont servi de base de comparaison à l'égard des simulations subséquentes faisant intervenir divers dispositifs antimise en charnière, et, grâce à l'information qui a pu être recueillie, elles ont permis d'approfondir les particularités de ces deux types d'instabilité dynamique.

C'est ainsi qu'il a pu être déterminé que les mises en charnière et les oscillations des semi-remorques surviennent lorsque les forces latérales que les pneus peuvent transmettre au sol sont dépassées, ou réduites, par les forces latérales induites aux points de contact pneus-chaussée. Il a été constaté que les oscillations des semi-remorques ne sont qu'une forme atténuée d'instabilité dynamique, étant donné qu'elles ont tendance à se résorber sans compensation aucune. Les chercheurs ont constaté la très forte influence de la configuration prise en compte sur l'amplitude de la mise en charnière ou des oscillations. Une augmentation des forces inertielles exercées sur le tracteur ou le porteur a tendance à atténuer l'amplitude de ces deux formes d'instabilité en raison de la résistance opposée par l'inertie mais en revanche, le moment de rotation autour de l'articulation est plus intense, et vice versa. L'amplitude des mouvements latéraux induits sera d'autant plus forte que la vitesse est élevée.

L'application des principes de la mécanique du solide libre a permis de calculer approximativement les moments de rotation autour de l'articulation à l'occasion d'une mise en charnière ou d'une oscillation. Dans le cas d'un tracteur à grand écartement tirant une semi-remorque vide de 14,7 m (48 pieds) de long, le moment de rotation calculé a été de l'ordre de 95 000 N.m lors d'une mise en charnière et de 140 000 N.m durant les oscillations de la semi-remorque. L'influence d'une augmentation progressive de la charge a été plus grande à ce chapitre lors des oscillations que lors d'une mise en charnière.

La détection des symptômes de mise en charnière et d'oscillations s'est révélée délicate étant donné que leurs caractéristiques respectives sont souvent difficiles à distinguer, du moins en régime transitoire, lorsque le véhicule effectue un virage normal. Le critère généralement admis pour détecter l'imminence d'une mise en charnière ou d'oscillations en circulation autoroutière est celui d'un angle d'articulation de 15 degrés; mais, il a été constaté qu'il s'agit d'un critère tardif. Il existe d'autres critères qui ont l'avantage d'être précoces, la vitesse angulaire de dérive - tracteur ou semi-remorque, par exemple - mais ils sont difficiles à mesurer. Il est possible aussi d'exploiter le rapport entre certaines caractéristiques ou certains paramètres du véhicule afin de pouvoir déterminer l'imminence d'un mouvement latéral induit, la vitesse angulaire de lacet et la vitesse, par

exemple. L'inconvénient de ces critères est qu'ils sont tributaires de la configuration représentée et qu'ils ne sont pas d'une utilité pratique étant donné les fréquentes permutations qui ont lieu entre une semi-remorque et le tracteur chargé de la tirer.

L'information issue de la recherche documentaire a permis de déceler les méthodes susceptibles de contrer les mouvements latéraux induits et leur efficacité a pu être approfondie grâce à des simulations sur ordinateur. Les méthodes utilisées ont été caractérisées selon l'effet produit : blocage, amortissement, bridage ou une combinaison de ces trois effets. Le blocage inhibe tout mouvement au-delà de l'angle amorçant un mouvement latéral réduit. L'amortissement qui peut prendre diverses formes ralentit les mouvements induits, permettant ainsi au conducteur de réagir à temps. Le bridage contient les mouvements induits à l'intérieur d'une plage donnée.

Quatre dispositifs ont été modélisés : constant torque (assimilé au blocage), Safe-T-King (bridage), Breeze (forme d'amortissement) et un dispositif à amortissement variable. Ils ont tous les quatre leurs avantages, ils ont été classés en fonction de leur efficacité à l'intérieur de la plage initiale de 1,5 seconde comme suit :

1. constant torque,
2. Safe-T-King,
3. dispositif à amortissement variable,
4. Breeze.

La plage de 1,5 seconde est une valeur choisie empiriquement comme représentative du temps de réaction type en conduite de nuit.

Ces dispositifs ont été peu ou pas efficaces à contrer les oscillations de la semi-remorque; en fait, la plupart ont eu un effet négatif. Cela s'explique par le fait que, lorsque l'articulation est rendue rigide, les valeurs élevées de moment de rotation induites par les oscillations font perdre aux roues des essieux moteurs leur capacité de guidage latéral, entraînant la mise en charnière de tout le véhicule.

Il est recommandé, suite à cette étude, que les recherches ultérieures se consacrent à distinguer les symptômes de mise en charnière de ceux des oscillations de la semi-remorque, et de ne s'attacher qu'à contrer les mises en charnière, et ce pour les raisons suivantes :

1. les dispositifs pour contrer les oscillations ont des effets plus nuisibles qu'autrement;

2. les oscillations de la semi-remorque ont tendance à se résorber sans compensation aucune;

3. les accidents attribuables aux mises en charnière comptent pour une plus grande proportion des accidents avec tués que les oscillations.

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NOMENCLATURE

α_i	Slip Angle of Tire i [ALPHA] (deg)
δ	Front Axle Steer Angle [DELTA] (deg)
Γ	Articulation Angle [GAMMA] (deg)
$\dot{\Gamma}$	Articulation Rate [GAMMA-DOT] (deg/sec)
$\ddot{\Gamma}$	Articulation Acceleration [GAMMA-DDOT] (deg/sec^2)
Ω	Understeer Gradient of Tractor
Ω_s	Understeer Gradient of Semitrailer
ψ	Yaw Angle of Tractor [PSI] (deg)
$\dot{\psi}$	Yaw Rate of Tractor [PSI-DOT] (deg/sec)
$\ddot{\psi}$	Yaw Acceleration of Tractor [PSI-DDOT] (deg/sec^2)
μ	General Friction Coefficient
a_y	Lateral Acceleration of Tractor (m/s^2)
AA	Distance Between Tractor Tandem Axles (m)
AAT	Distance Between Semitrailer Tandem Axles (m)
A1	Distance From Tractor CG to Front Axle (m)
A2	Distance From Tractor CG to Rear Axle (m)
A3	Distance From Trailer CG to Fifth Wheel (m)
A4	Distance From Semitrailer CG to Rear Axle (m)
BB	Distance From Tractor Rear Axle to Fifth Wheel (m)
CALF	Cornering Stiffness of Tires (N/deg)
CG	Centre of Gravity
$f(\alpha_i, \bar{F}_{zi})$	Normalize Lateral Force Characteristic (ratio of F_{yi} to $\bar{F}_{zi}$)
F_{xi}	Longitudinal Force on Tire i
F_{yi}	Lateral Force on Tire i
F_{zi}	Normal Load on Tire i (N)
$\bar{F}_{zi}$	Static Normal Load on Tire i (N)
F(S)	Slip Rolloff Function
FSX	Attempted Brake Force on One Side of an Axle (N)
GAM1	Portion of Lateral Load Transfer on Front Axle of Tractor
GAM3	Tractor Tandem Axle Load Transfer Coefficient
GAM4	Semitrailer Tandem Axle Load Transfer Coefficient
g	Acceleration due to gravity (9.82 m/sec^2)

I_s	Yaw Moment of Inertia of Semitrailer [ITZZ] ($kg \cdot m^2$)
I_t	Yaw Moment of Inertia of Tractor [IZZ] ($kg \cdot m^2$)
IQUIT	Maximum Articulation Angle Obtained in Program (deg)
L	Tractor Wheelbase (m)
L_s	Semitrailer Wheelbase (m)
M_s	Mass of Semitrailer [GVW2] (kg)
M_t	Mass of Tractor [GVW1] (kg)
MUMOD	Modified Longitudinal Friction Coefficient Due to Braking
MUS	Sliding Friction Coefficient
MUP	Peak Friction Coefficient
MU5	Fifth Wheel Friction Coefficient
R	Turning Radius (m)
r	Yaw Rate (deg/sec)
RAD5	Equivalent Radius of Fifth Wheel (m)
S	Slip (0% to 100%)
SP	Slip Corresponding to MUP
T	Moment Due to Friction at Fifth Wheel (N·m)
TIMF	Maximum Simulation Time (sec)
TJ	Anti-Jackknife Device Torque (N·m)
TRA1	Half the Lateral Distance Between Centres of Tire Contact on Tractor Front Axles (m)
TRA2	Half the Lateral Distance Between Centres of Tire Contact on Tractor Rear Axles (m)
TRA3	Half the Lateral Distance Between Centres of Tire Contact on Semitrailer Axles (m)
U-vel	Longitudinal Velocity of Tractor (km/hr)
U_c	Critical Velocity of Tractor [$U_c = (Lg/\Omega)^{\frac{1}{2}}$]
U_{cs}	Critical Velocity of Semitrailer [$U_{cs} = (L_s g/\Omega_s)^{\frac{1}{2}}$]
V-vel	Lateral Velocity of Tractor (km/hr)
VEL	Initial Velocity of Vehicle [U-direction] (km/hr)
$\bar{V}$	Absolute Velocity of Tractor (km/hr)
Z0	Height of Fifth Wheel Above Ground (m)
Z1	Height of Tractor CG Above Ground (m)
Z2	Height of Semitrailer CG Above Ground (m)

1.0 INTRODUCTION

In October of 1988, T E S Limited was contracted by the Transportation Development Centre, Transport Canada, to conduct a feasibility study of an anti-jackknifing system (AJS) for tractor semitrailers. The objective of the system was to control articulation of the vehicle during jackknife and trailer swing, independent of actions by the driver.

This report includes a summary of statistical information on jackknife accidents, a review of the design and performance of existing AJS patents and products and a summary of the experiences of users of jackknife control devices.

The results of computer simulations which were used to determine the characteristics of jackknife and trailer swing of four distinct vehicle configurations are presented. Also presented are the results of computer analysis of the performance of anti-jackknifing devices with regard to vehicle stabilization and the effects of control parameter variations.

Finally, conclusions and recommendations are provided regarding the technical feasibility of an effective AJS for various tractor semitrailer configurations and trailer load conditions.

At various times during the study, the findings of the research group were presented to the Steering Committee, which was composed of Canadian authorities in heavy trucking (See Acknowledgements for a list of the members). The comments and suggestions of the Steering Committee were used to guide the project.

2.0 BACKGROUND

Highway accidents involving large articulated vehicles cause a significant amount of damage and are often accompanied by personal injuries and fatalities. Such accidents attract a great deal of public attention largely due to the resulting traffic problems. Consequently, many groups have promulgated the idea of using safety devices to help prevent these catastrophic accidents. As a result, devices such as anti-lock brake systems and articulation control mechanisms have been developed to inhibit the phenomenon known as jackknifing.

Jackknifing occurs when the wheels of the tractor or semitrailer lose traction and allow either section of the articulated vehicle to rotate out of control about the fifth wheel king pin. There are two basic modes of jackknifing. The first, referred to as a tractor jackknife and explained in the simplest of terms, occurs when the trailer remains stable on the roadway and the tractor rotates about the king pin in a highly unstable fashion. The second, referred to as a trailer swing, is essentially the reverse. In this case, the tractor is stable and the trailer swings about the king pin.

A tractor jackknife commonly results from excessive braking and usually subsequent wheel lock which eliminates the directional stability of the locked rear wheels of the tractor. In this condition, the rear axles of the tractor are pushed sideways from the direction of motion due to the momentum of the tractor and semitrailer. This form of instability has been the cause of many fatal accidents due to the abrupt manner in which the articulated vehicle sweeps across the path of oncoming traffic. This type of jackknife can be alleviated to a degree by reducing or eliminating the tire side forces at the steering wheels of the tractor at the same time that lateral traction is lost at the rear wheels (i.e. braking at the front steering axle). In some cases, this is sufficient to avoid a jackknife but will still result in the tractor and semitrailer both sliding forward out of control in the general direction of travel. Anti-lock Brake Systems (ABS) for this reason have become an attractive option on combination vehicles.

A tractor jackknife can also be caused by actions which produce excessive acceleration at the rear wheels, or the cutting in of an engine retarder on icy surfaces. Although this condition, known as a power jackknife, can be equally as dangerous, it is generally not as frequent except in mountainous or hilly terrain and/or in adverse weather conditions.

A trailer swing occurs when the trailer swings away from the direction of travel. If the articulation angle becomes acute and the tractor and trailer physically interfere, the trailer will sweep across several lanes of traffic or the tractor trailer combination will exit the roadway out of control. This is a common scenario when traction loss occurs during downhill braking. In this mode of instability, the driver usually retains control of the tractor while the trailer swings out of control. In either of the trailer swing instabilities, operation of the vehicle brakes tends to aggravate rather than alleviate the tendency to jackknife.

Over the years, a number of devices have been developed to prevent jackknifing. In some cases limited performance and excessive cost have inhibited their acceptance by the trucking industry. Other devices simply have not been tested.

3.0 METHODOLOGY

The primary purpose of this study was to determine the feasibility of an anti-jackknife system (AJS), for various tractor semitrailer loading configurations. The proposed system would control articulation of the vehicle during jackknife and trailer swing, independent of the actions of the driver and would meet industry performance requirements. In order to achieve this objective, several tasks were identified, see Figure 1.

A review of Canadian and U.S. statistics concerning jackknife accident/incidents involving tractor semitrailers was performed. Statistical information was obtained from federal transportation departments, provincial insurance agencies, and several university affiliations such as the Transport Research Institute, at the University of Michigan. See Appendix "A" for a complete list of contacts.

World-wide information on existing and patented anti-jackknife devices was surveyed. AJS design features and performance capabilities were reviewed. In addition, the experience of users of AJS was documented and general comments from the trucking industry were obtained. Trucking associations, truck manufacturers/dealers, fleet managers and government agencies were solicited for comments on this issue.

The aforementioned background research assisted in the formulation of the conditions that lead to jackknifing or swing and the essential requirements of an AJS for acceptance by the trucking industry.

A suitable computer software package was selected for the analysis of articulated vehicle dynamics and anti-jackknife control performance. Four vehicle configurations were selected to give a wide range of tractor semi-trailer combinations. Steady state characteristics for each configuration were determined. Two highway manoeuvres, namely a constant radius turn and a lane change, were used to evaluate the AJS devices.

From the baseline runs performed, characteristics of jackknife and trailer swing were determined. In addition, methods for sensing the onset of both jackknife and trailer swing were investigated.

During the AJS design synthesis, the results of the computer simulations were compared to the results of the theoretical analysis and the driver response parameters. This work included an evaluation of the methods of achieving control of jackknife and trailer swing. In addition, the influence of ABS and front axle brakes on the results obtained with the application of an AJS was investigated.

Finally, conclusions drawn from the study and recommendations for further work in this area were prepared.

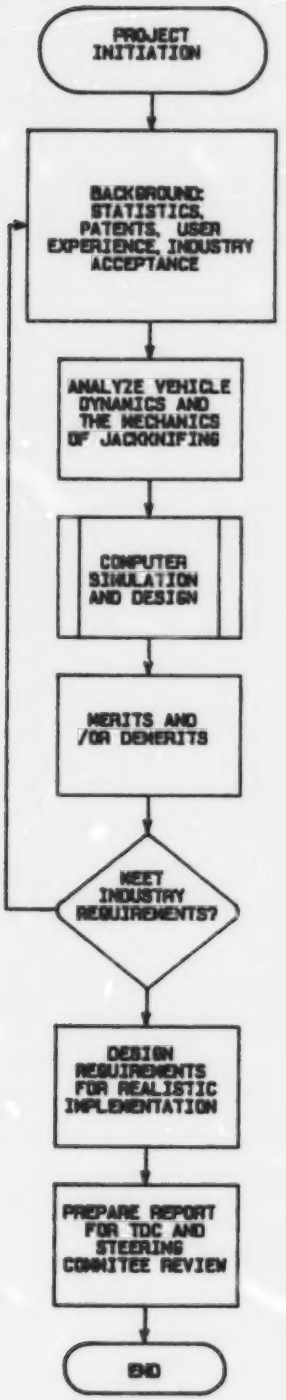

PROJECT INITIATION
BACKGROUND: STATISTICS, PATENTS, USER EXPERIENCE, INDUSTRY ACCEPTANCE
ANALYZE VEHICLE DYNAMICS AND THE MECHANICS OF JACKKNIFING
COMPUTER SIMULATION AND DESIGN
MERITS AND /OR DEMERITS
MEET INDUSTRY REQUIREMENTS?
DESIGN REQUIREMENTS FOR REALISTIC IMPLEMENTATION
PREPARE REPORT FOR TDC AND STEERING COMMITEE REVIEW
END

Figure 1 Project Methodology

4.0 SURVEY OF JACKKNIFE STATISTICS, PATENTS/PRODUCTS AND INDUSTRY REQUIREMENTS

The results of the surveys, jackknife* statistics, patent/product information, and the comments of users of AJS devices are presented in this section. From the information obtained, conditions that lead to jackknifing and trailer swing were classified and essential requirements of an AJS were highlighted.

4.1 Statistical Information on Jackknifing Accidents

A review of statistics pertaining to jackknifing accidents was conducted at the onset of the program. The purpose of this review was to determine the extent and cost of the problem, as well as to classify the conditions that lead to tractor jackknife or trailer swing.

To obtain the required statistical data, a number of government organizations, insurance companies, trucking associations, universities, and libraries, in both Canada and the U.S.A., were contacted (see Appendix "A" for a complete list of sources contacted).

It was determined that detailed statistical information on jackknifing was generally unavailable and what was available was somewhat suspect. This was considered to be because police officers were often unable to distinguish between an accident due to jackknifing and jackknifing due to an accident (which are often referred to as 1st and 2nd events, respectively). As well, the accident forms used by police officers were often structured so that only a minimum amount of information was obtained. In addition to this, the information had generally not been compiled into a statistical data base pertaining to jackknifing. A limited number of studies in Canada and the U.S.A. existed that included a more thorough investigation and statistical compilation of data. One example was the "Commercial Vehicle Accident Survey" conducted by the Ministry of Transportation of Ontario in 1982 (1). The findings presented in this report were based mainly on these studies.

It was determined that, in Canada and the U.S.A., combination trucks (including tractor and semi-trailer, tractor and multiple trailer, straight truck and full trailer) represent approximately 1% of the total vehicles on the road. However, they account for about 6% of the total vehicle miles travelled. When comparing the amount of accidents, combination trucks have a better record than other vehicles on the road as they account for only 3% of total vehicle accidents. When comparing the number of accidents involving fatalities, combination trucks have a worse record than other vehicles, as they account for approximately 9% of total fatalities (see Tables 1 and 2 for more details on these statistics).

NOTE:* Unless otherwise specified, the term jackknife in this section is used to represent both tractor jackknife and trailer swing.

It appears that 5 to 10% of combination truck accidents are jackknife or trailer swing accidents. Some studies, such as the one conducted by the Insurance Institute for Highway Safety (2) in the U.S.A., listed much higher percentages which were in the order of 28% to 55%. Examination of fatal jackknifing accidents as a percentage of total jackknifing accidents showed a range of 0 to 10% for the Canadian provinces where statistical information was available.

Accurate cost figures for these accidents was not available. However, one report estimated that combination truck accidents averaged about $15,000 per incident.

Examination of conditions that lead to jackknifing showed that approximately 70 to 80% of vehicles involved in jackknife or trailer swing were empty or lightly loaded. As well, 70 to 90% of jackknifing or trailer swing accidents occurred on slippery roads (ice, snow, rain).

Table 1 General Statistics for Combination Units*

	CANADA	U.S.A.
Number of Units	(90,000)	1,260,000**
$\frac{\text{Total Combination Units}}{\text{Total No. of Vehicles}} \times 100$	(1.0%)	0.9%
$\frac{\text{Total Miles Travelled}}{\text{Total Vehicle Miles Travelled}} \times 100$	(6)	5.6%
Total Accidents	(20,000)	219,156**
$\frac{\text{Total Accidents}}{\text{Total Vehicle Accidents}} \times 100$	1.8%	3%
Total Fatal Accidents	(300)	4,232**
$\frac{\text{Total Fatal Accidents}}{\text{Total Vehicle Fatal Accidents}} \times 100$	6.1%	9%

NOTES:

*Combination Units include: tractor and semi trailer, tractor and multiple trailer, straight truck and full trailer.

**Heavy Truck Safety Study, NHTSA, 1984(3)

( ) represent estimates

Table 2 Jackknifing Statistics for Combination* Units

	CANADA	U.S.A.
$\frac{\text{Jackknifing Accidents}}{\text{Total Accidents}} \times 100$	7.8% B.C. 1987 5.8% Sask 1985 10.5% Quebec 1982-86	3.2% Texas, 1981 8.5% UMTRI
$\frac{\text{Fatal Jackknifing Accidents}}{\text{Fatal Accidents for all Trucks >10,000 lbs}} \times 100$		ǂ 5.7% 1st Event ǂ 3.8% 2nd Event
$\frac{\text{Fatal Jackknifing Accidents}}{\text{Jackknifing Accidents}} \times 100$	8.7% B.C. 1987 9.6% Quebec 1982-88 0% Saskatchewan 1987	(8%)

NOTES:

*Combination Units include: tractor & semitrailer, tractor and multiple trailer, straight truck full trailer.

ǂ Trucks Involved in Fatal Accidents, 1980-84, UMTRI (4)

( ) represent estimates

4.2 Review of AJS Patents and Products

To establish which AJS designs had been developed to date, a patent search for Canada and the U.S.A. was undertaken. Due to the large number of patents closely or partly related to this topic, the patent search approach endeavoured to select a sample of patents which was representative of the type of designs which were available, rather than perform a full review of all possible patents. In addition to this patent search, other sources such as SAE literature, libraries and trucking firms were consulted to determine if additional AJS designs existed.

The information on these devices was reviewed and the devices were categorized as follows: devices independent of the fifth wheel, devices about the fifth wheel and other devices (see Appendix "B").

The objective of this review was to establish the advantages and disadvantages of each device and to determine whether they merited further evaluation by computer simulation. As well, it was intended that the positive design features of these devices be combined or used to create more effective AJS. Those devices which appeared to have merit are presented in Figures 2, 3, 4 and 5.

4.3 Survey of AJS Users

A survey of individuals involved in the trucking industry was conducted. The purpose of the survey was to solicit opinions and comments regarding both the jackknife phenomenon and anti-jackknifing devices. Persons contacted in the survey included truck drivers, fleet managers, members of trucking associations and representatives of tractor and trailer manufacturers (see Contact List (Appendix "A")). The survey was conducted primarily in Canada, but also included contacts in the U.S.A., the United Kingdom and Europe. This section first presents opinions related to jackknifing offered by people involved in the trucking industry and then outlines opinions held by users of anti-jackknifing devices.

There were many opinions regarding jackknifing and AJS encountered throughout the survey, however, a number of common themes were noticed. People involved in the trucking industry were generally familiar with the phenomenon of jackknifing but very few seemed to have a good understanding of the underlying principles that cause a jackknife.

Those surveyed were generally aware of the existence of AJS devices, but this awareness was usually from "hearsay" and not from actual use. One of the most recurring comments offered by the survey participants was that AJS had not gained acceptance by the industry. In general, for those who were aware of AJS but did not own any systems, their lack of interest in obtaining AJS was due to the following factors: extra expense, awkwardness in use, concern with tractor/trailer compatibility, and uncertainty as to whether the device really works. Further, the conservative attitude of the trucking industry itself was mentioned as a reason for the non-acceptance of AJS. Because they were constantly being offered new add-on devices, truckers had difficulty

knowing what was valid and where to draw the line on purchases. Thus truckers tended to be sceptical of new products.

From the fleet operators perspective, trucking is a very competitive business. The vigorous competition promotes cost cutting wherever possible and left little capital available for safety issues.

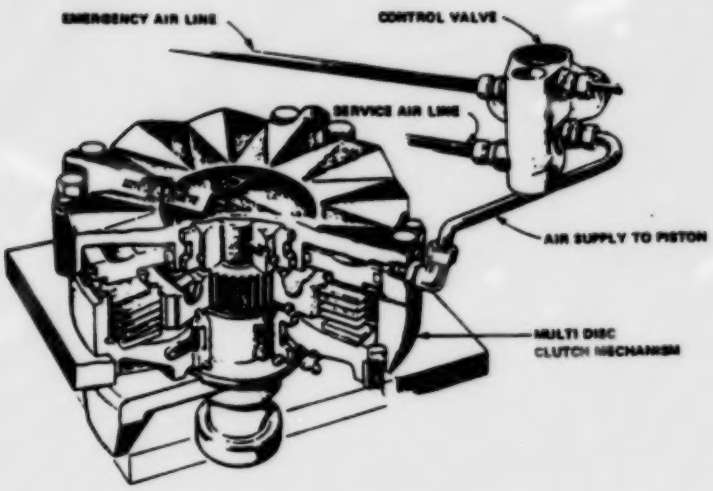
Several fleet operators involved in the transport of hazardous goods indicated that jackknifing was a definite concern and they were very much interested in the implementation of a failsafe AJS.

Several users of AJS were approached for information. One user operated a petroleum haulage operation. They had 5 trailers in total, of which 3 were equipped with Breeze systems and the remaining 2 were planned to be equipped with similar units. They were pleased with the operation and reported no occurrences of jackknife.

Another operator had a fleet of 17 units of which 15 were equipped with Breeze AJS units. He became interested in the Breeze Unit after seeing it in use at a Southern U.S. truck driver training centre. He was very pleased with the performance of the units and cites statistics to support this. He stated that prior to the installation of the Breeze units his fleet was experiencing approximately four jackknife incidents per fleet-year. After installation the fleet had no occurrences of jackknife. In one situation he believed the unit prevented two deaths.

Another satisfied user of the Breeze device operated a service for the New York Times newspaper and had 60 AJS units in total. This operator became interested in AJS after winning a free unit in a raffle. After trying one, the decision was made to go ahead and purchase 59 more. Again, the operator reported having roughly 5 jackknife incidents per year before installing the breeze units and had approximately 1 jackknife per year now. He expressed some doubt that the AJS device was always in operation (operator forgot or neglected to attach cables to tractor).

Device: HOPE (6)



Description & Operation: A multi-disk slipping clutch mounted to the trailer, with a forged king pin arm which fits into the Vee of the 5th wheel. Activated by air pressure when vehicle brakes applied.

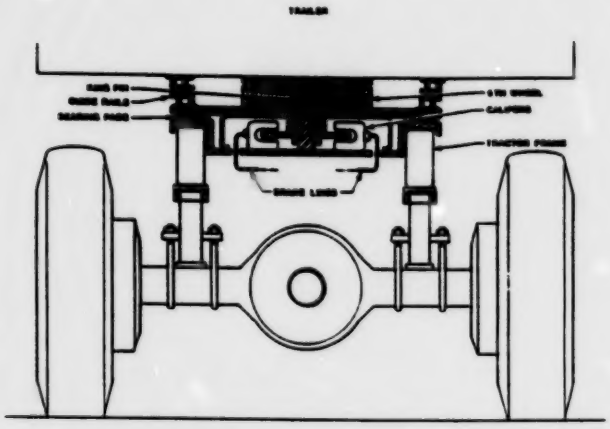
Torque Capability: Max. Torque = 6,000 N•m

Pros:
- Modifications to tractor are not required
- Sealed disks tend to eliminate moisture problems

Cons:
- Braking Capacity (Resistive torque) is sufficient (see analysis Section 6.3 or MTO Report (1))
- Tractor/trailer is restricted from returning to the inline position

Figure 2 Patented Friction Devices

Device: BRAKE CALIPER & DISK (MacKay Interface)



Description & Operation: A disk, connected to the trailer at the 5th wheel, is braked by hydraulically actuated calipers connected to the tractor.

Torque Capability: Depends on disk size and quantity of calipers

For a "drop down and bolt on" assembly, the maximum disk diameter is 58 cm. For existing calipers the largest torque capacity is:

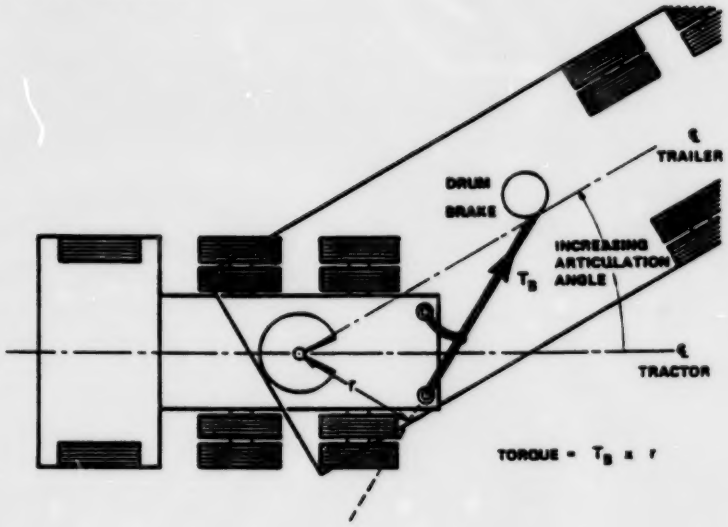
Two Calipers - Torque = 19,000 N•m
Three Calipers - Torque = 28,500 N•m
Four Calipers - Torque = 38,000 N•m

Pros: Resistive torques can be generated that are quite high.

Cons: The cost to achieve the levels of torque required is high (see Analysis Section 6.3)

Figure 2 Patented Friction Devices (continued)

Device: BREEZE



Description & Operation: A drum brake mounted on the trailer connects to the tractor by means of cables.

The Drum brake is activated by air pressure when the vehicle brakes are applied. Resistive moment increases with articulation angle.

Torque Capability: At an articulation angle of:

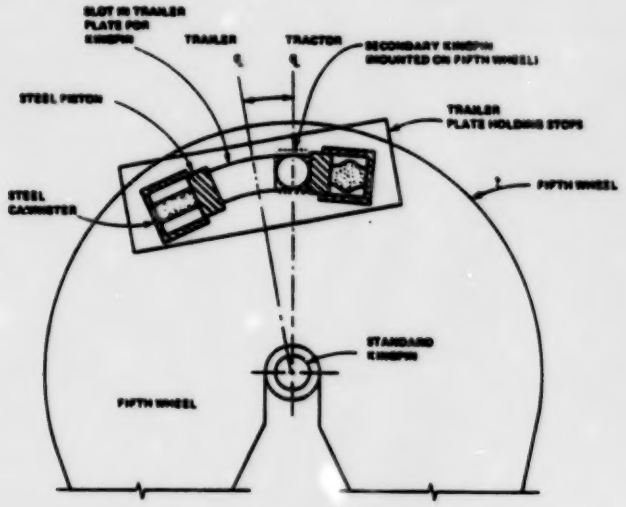
6°	Torque = 19,300 N•m
10°	Torque = 28,400 N•m
18°	Torque = 34,000 N•m

Pros:
- Large moment arm reduces the required AJS brake force to obtain high torques
- Relatively large resistive torques are possible.
- tractor/trailer can return to inline position without any system resistance.

Cons:
- All trailer types are not adaptable
- Driver must remember to connect cables
- The device is activated/actuated by vehicle brake application.

Figure 2 Patented Friction Devices (continued)

Device: SAFE-T-KING



Description & Operation:
A pin mounted in the fifth wheel extends upward, when activated by the driver, and engages with a slot in a plate mounted to the trailer. The slot limits articulation to ± 16°.

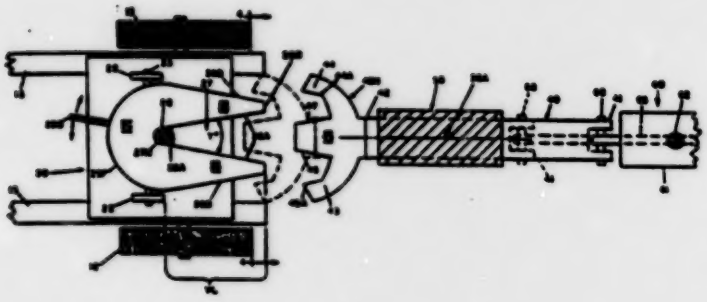
Torque Capability: At ± 16°, torque capacity is determined by the structural strength of the assembly.
Damped from 12° to 16° (0 to 11,000 N•m)

Pros:
- It is simple and inexpensive.
- Jackknife is prevented from proceeding to the point where the tractor rotates into the trailer.
- Sensing is not required.

Cons:
- Tractor may rotate out of control during trailer swing.
- No significant resistance to jackknifes or swings during articulation to ± 16° (vehicle will be partially in neighbouring lanes).
- Fifth wheel is used to support pin (5th wheel manufacturers will not condone this).

Figure 3 Patented Articulation Limiting Devices

Device: RED ARROW



Description & Operation:
A bar or fork engages with the Vee on the 5th wheel, when activated by the driver, thereby limiting the articulation to ± 16°.

All other characteristics will be approximately the same as the SAFE-T-KING.

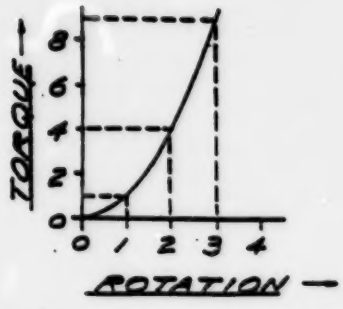
Torque Capability: At ± 16°, torque capacity is determined by the structural strength of the assembly.
Damped from 12° to 16° (0 to 11,000 N•m)

Pros:
- It is simple and inexpensive.
- Jackknife is prevented from proceeding to the point where the tractor rotates into the trailer.
- Sensing is not required.

Cons:
- Tractor may rotate out of control during trailer swing.
- No significant resistance to jackknifes or swings during articulation to ± 16° (vehicle will be partially in neighbouring lanes).
- Fifth wheel is used to support the bar or fork (5th wheel manufacturers will not condone this).

Figure 3 Patented Articulation Limiting Devices (continued)

Device: MATHER



Description & Operation: Fluid flow, induced by articulation, is throttled creating a resistive moment to the articulation proportional to the articulation rate squared.

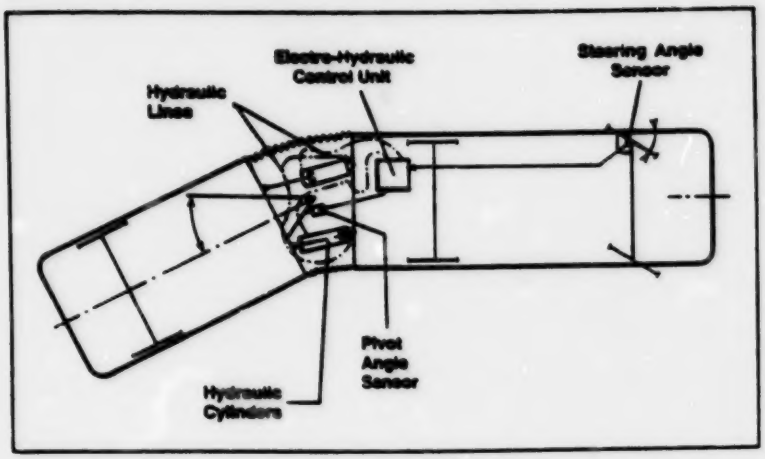
Torque Capability: 5,600 N•m

Pros:
- Little resistance is generated at low articulation rates (i.e. steering)
- Large resistance is generated at high articulation rates (i.e. jackknifing).
- Sensing is not required to activate the device.

Cons:
- A jackknife or swing is not stopped but the driver has more time to react.
- Little resistance is generated due to low articulation rate for trailer swing.
- Dampening coefficient is a compromise between not impeding steerability and providing sufficient resistance to slow articulation during jackknife.
- Resistive torque is too low due to sealing limitations.

Figure 4 Patented Fluid Dampening and Locking Devices

Device: HYDRAULIC CYLINDERS



Description & Operation: Hydraulic cylinders linked to the articulation are locked or throttled when jackknife or swing is sensed. (Picture shows articulated bus with this type of system).

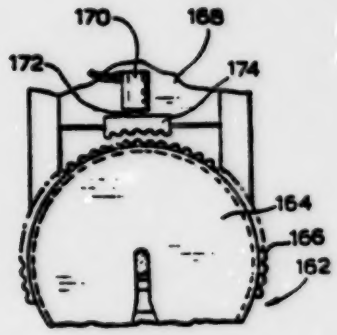
Torque Capability: Depends on cylinder size, pressure capacity and moment arm. Using a 23 cm moment arm and two 10 cm bore hydraulic cylinders rated for 437 KPa max. pressure.
Torque = 64,400 N•m

Pros:
- Large resistive torques are generated.
- Hydraulic systems are adaptable to control options.

Cons:
- More space is required than for other types of devices.

Figure 4 Patented Fluid Dampening and Locking Devices (continued)

Device: Patent No. 1034 618 (Can)



Description & Operation: Toothed ram on trailer forced against gear fixed to 5th wheel.

Torque Capability: Limited by the strength of the components & assembly structure.

Pros:
- Locking at any articulation angle may prevent tractor or trailer from encroaching onto neighbouring lanes during jackknife or swing.

Cons:
- Concept has never been verified
- Locking is not desireable if steering change is required
- Locking may not be desireable for trailer swing

Figure 5 Patented Articulation Locking Devices

4.4 General Conditions that Lead to Jackknifing

Based on the jackknifing statistics, trucking industry survey and review of relevant literature, general conditions that lead to jackknifing were assembled.

The information gathered indicated that the most common forms of jackknifing were due to locking of the wheels (either the tractor rear tires or the semitrailer rear tires) or spinning of the tractor drive wheels. Jackknifing could also be caused by excessive cornering speed.

The most obvious cause of wheel lock was excessive brake application or selection of too low a gear on the downhill descent. Likewise, spinning of the tractor drive wheels was caused by excessive throttle application. However, there were a number of conditions or factors which could promote a more aggressive jackknife. These factors were classified as highway/environmental and vehicle.

Highway/environmental factors can be separated into the following categories:

- Road surface conditions (affect coefficient of friction)
    - dry
    - wet (dew, rain)
    - icy
    - snow covered
- Road Geometry
    - banked curve
    - sharp curve
    - camber
    - slope
- Wind

The vehicle factors can be separated under:

- Cargo Loading
    - payload
    - load distribution: fore/aft and side/side
- Unbalanced brake forces between wheels
- Improperly adjusted brakes
- Braking time delay
- Fifth wheel location
- Vehicle aerodynamic properties
- Vehicle stability
- Vehicle geometry and weight.

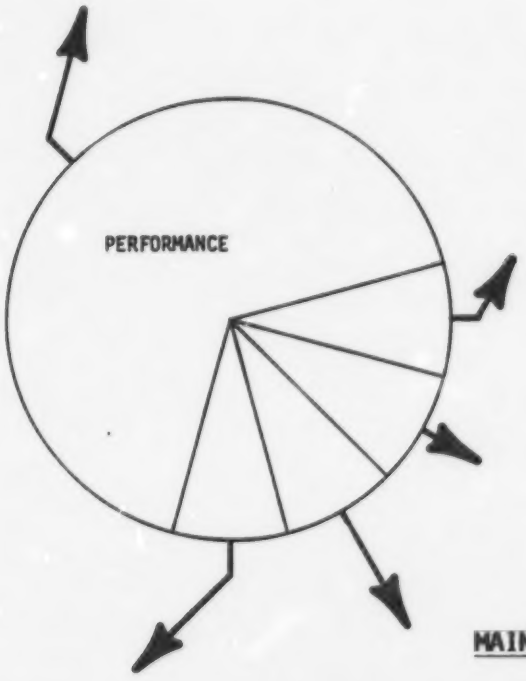
One of the most important vehicle factors affecting jackknife is trailer payload. The vehicle brakes were designed to stop a fully loaded vehicle. A loaded trailer can weigh up to five times the weight of an empty trailer. The removal of this load could cause the wheels to lock under light brake applications.

4.5 Industry Requirements of an AJS

From the results of the surveys presented in sub-sections 4.1 through 4.4, industry requirements of an AJS were developed. The requirements were classified into the following five generic groups: performance, operation, installation, maintenance and cost. The pie-chart below illustrates the relative importance of the groups.

PERFORMANCE

- Prevent jackknifing and trailer swing under all conditions
- Activate automatically and independent of driver
- Adapt to all tractor-trailer and fifth wheel configurations
- Maintain driver controlability
- Introduce no other hazards as a result of normal use or malfunction



OPERATION

- No interference with normal operation of vehicles
- Minimal hookup time

INSTALLATION

- Easy to retrofit
- Ideally contained totally, on tractor or totally on trailer

MAINTENANCE

- Easy to maintain

COST

- Reasonable ($5000 maximum)

5.0 COMPUTER SIMULATION

5.1 Methodology

An illustration of the approach taken toward analysis of the jackknife problem through computer simulation is depicted in Figure 6. A logical starting point was to first determine suitable software availability, and then select the package that was best suited to our purpose of analyzing the jackknife problem (refer to Section 5.2 and Appendix "D" for details).

Once suitable software had been selected, the vehicle configurations, manoeuvres and environmental conditions (friction values) to study jackknifing were determined (see Section 5.3).

The simulation process required that the steady-state handling characteristics of the vehicle configurations under study be determined. These characteristics indicated the general handling tendencies of the combinations (i.e. naturally oversteer, understeer or neutral steer). They also aided in the prediction of manoeuvre outcome based on the results of simulation runs with other configurations. Steer inputs to the program for a particular configuration were derived from the corresponding handling diagram (the combined plots of "Lateral Acceleration vs L/R" and "Lateral Acceleration Versus RL/U-δ"), Appendix E.

A baseline for comparison was required. This was accomplished by successfully running the vehicles through the manoeuvres without inducing jackknife or trailer swing. When jackknife and trailer swing were induced, for each configuration and each manoeuvre, the results for that group of simulations were compared with the baseline runs to determine the characteristics of both jackknife and trailer swing (Section 5.4).

Design of an anti-jackknife system (AJS) based on the determined characteristics of jackknife and trailer swing was initiated. The device was first tested on jackknife and trailer swing induced simulations for a constant radius turn. The results were studied and the merits of the device were then determined. If the device was deemed successful it would be applied to the jackknife and trailer swing induced simulations of a lane change. If it was deemed unsuccessful the design would be modified and the process repeated (depicted by the first decision block in Figure 6). Similarly if a device proved successful in preventing both jackknife and trailer swing in the constant radius turn and failed in the lane change (second decision block) it would be modified and re-tested until the device successfully improved the vehicle handling through both manoeuvres. A final check was done by determining the effect of the device when combined with ABS and front axle braking (the final decision block).

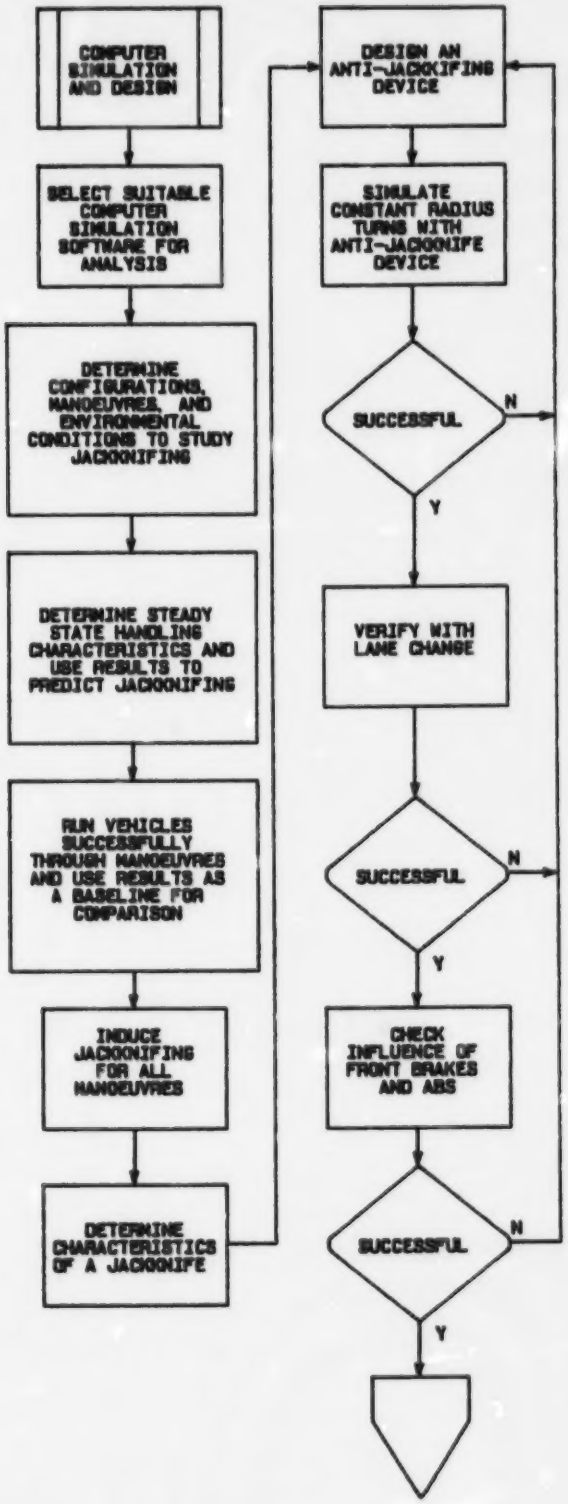

COMPUTER SIMULATION AND DESIGN
SELECT SUITABLE COMPUTER SIMULATION SOFTWARE FOR ANALYSIS
DETERMINE CONFIGURATIONS, MANOEUVRES, AND ENVIRONMENTAL CONDITIONS TO STUDY JACKKNIFING
DETERMINE STEADY STATE HANDLING CHARACTERISTICS AND USE RESULTS TO PREDICT JACKKNIFING
RUN VEHICLES SUCCESSFULLY THROUGH MANOEUVRES AND USE RESULTS AS A BASELINE FOR COMPARISON
INDUCE JACKKNIFING FOR ALL MANOEUVRES
DETERMINE CHARACTERISTICS OF A JACKKNIFE
DESIGN AN ANTI-JACKKIFING DEVICE
SIMULATE CONSTANT RADIUS TURNS WITH ANTI-JACKKNIFE DEVICE
SUCCESSFUL
N
Y
VERIFY WITH LANE CHANGE
SUCCESSFUL
N
Y
CHECK INFLUENCE OF FRONT BRAKES AND ABS
SUCCESSFUL
N
Y

Figure 6 Computer Simulated Methodology

5.2 Selection of Software

There are a number of documented software packages related to heavy articulated vehicles currently available. The problem at hand was to select a package suitable to analyze the jackknife problem. Listed below are names, origins and purposes of software packages that surfaced during the course of the project. This list is intended for reference purposes only and is by no means a comprehensive list of available packages.

1. Name: "Simulation Programme for Tractor Semi-trailer Vehicles Equipped with a Hydraulic Type Anti-Jackknifing Articulation Device" (6)
   Origin: University of Sherbrooke; P. Bourassa, A. Laneville, G. Payre, L. Maillette
   Purpose: 4 D.O.F. Dynamic Model for Tractor Trailers (Braking and an AJS device incorporated).

2. Name: "Simulation of Tractor-Semitrailer Handling" (7)
   Origin: Cornell University; A.I. Krauler, R.K. Wilson
   Purpose: 8 D.O.F. Dynamic Model based on the equations of Mikulcik modified to incorporate tire forces and wheel dynamics.

3. Name: "Handling Response and Accident Reconstruction of Tractor Semitrailers Using Micro and Mini computers" (8)
   Origin: J.E. Martinez (Texas A & M University), J.G. Wallingford (Vertifact Corp.)
   Purpose: Program is geared toward accident reconstruction using a 4 D.O.F. Model.

4. Name: "Linear Yaw Plane Model" (9)
   Origin: University of Michigan Transport Research Institute (UMTRI)
   Purpose: To study directional behaviour of multiple articulated vehicles at a constant velocity.

5. Name: "Yaw/Roll Model" (9)
   Origin: University of Michigan Research Institute (UMTRI)
   Purpose: Developed for purposes of predicting the directional and roll response of single and multiple articulated vehicles engaged in steering manoeuvres which approach the roll-over conditions (constant velocity is assumed).

6. Name: "Phase IV Model" (9)
   Origin: University of Michigan Research Institute (UMTRI)
   Purpose: A complex model used to simulate the braking and steering dynamics of multiple articulated vehicles (a maximum of 71 D.O.F. may be incorporated).

7. Name: "HSRI/MTC Simulation Program for a Tractor-Trailer in the Horizontal Plane (10)

Origin: Highway Safety Research Institute (HSRI), University of Michigan. Modified by the Ontario Ministry of Transportation and Communications.

Purpose: To study the Vehicle Dynamics of Tractor Semi-Trailers with or without fifth wheel anti-jackknife devices.

It should be noted that simulation models 4, 5 and 6 were obtained from UMTRI by the Roads and Transport Association of Canada (RTAC) as part of their "Vehicle Weights and Dimensions Study". These models were installed on the VAX at the Systems Laboratory of the National Research Council in Ottawa. A user's guide (9) was developed by two visiting researchers, Dr. Wong and Dr. El-Gindy, with a view to making these analytical tools available to the Canadian transportation and research community at large.

Simulation model 7 (described in Appendix "D") was chosen to be best suited for the study of jackknife based on a number of factors:

- Both brakes and steer inputs were incorporated,
- ABS and AJS device models were developed and incorporated in the program,
- both the executable code and the source code was easily attainable (from MTO), and
- most importantly, the program had been successfuly validated against actual test data.

Other models, such as models 1 and 6, could have been used, but fell short of meeting all the requirements. For example, Model 1 from the University of Sherbrooke did not incorporate tandem axles, and the brake and tire models coded were over simplified for the purpose of studying jackknifing. Model 6, confirmed by the Steering Committee, was too complicated for the purpose at hand and required many inputs that demanded experimental measurements.

5.3 Vehicle Configurations and Manoeuvres

The vehicle configurations and manoeuvres chosen to conduct this study closely represented the typical range in possible tractor semitrailers on the roads today. The actual numbers presented were the result of meetings, comments and suggestions from members of the Steering Committee.

5.3.1 Vehicle Parameters

Two tractors and two trailers were chosen for simulation analysis. Numeric parameters of these vehicles are presented in Table 3, Table 4 and Table 5. If raw data could not be found for a particular component, estimates were made using information found in "The Factbook" (11).

Table 3 Tractor Data (3 axle)

DESCRIPTION	VALUE*	
	Tractor 1	Tractor 2
Wheelbase (m) [L]	5.6	3.6
Weight (kg) [GVW1]	9,047	6,695
Yaw Moment of Inertia (kg-m^2) [IZZ]	62,000	19,978
Dist. Between Tandem Axles (m) [AA]	1.52	1.52
Dist. from CG to Front Axle (m) [A1]	2.565	1.666
Dist. from CG to Rear Axle (m) [A2]	3.025	1.966
Dist. from CG to Ground (m) [Z1]	1.118	1.118
Dist. from Rear Axle to Fifth Wheel (m) [BB]	0.1524	0.1524
Height of Fifth Wheel Above Ground (m) [ZO]	1.32	1.32
Half Lateral Dist. Between Centres of Front Tire Contact (m) [TRA1]	1.029	1.029
Half Lateral Dist. Between Centres of Rear Tire Contact (m) [TRA2]	0.991	0.991
Portion of Total Lateral Load Transfer on Front Axle [GAM1]	0.110	0.110
Tandem Axle Load Transfer Coef. [GAM3]	0.15	0.15

*Derived from reference (11)

Table 4 Trailer Data (2 axle)

DESCRIPTION	VALUE*	
	Trailer 1	Trailer 2
Length (m)	14.6	8.2
Weight (kg) [GVWZ]	6,260 (empty) 19,867 (loaded)**	2,948 (empty) 16,555 (loaded)**
Trailer Moment of Inertia (kg-m^2) [ITZZ]	150,194 (empty) 399,820 (loaded)	46,927 (empty) 130,522 (loaded)
Dist. Between Tandem Axles (m) [AAT]	1.52	1.52
Dist. from CG to Ground (m) [Z1]	1.65 (empty) 1.83 (loaded)	1.65 (empty) 1.83 (loaded)
Dist. from CG to Fifth Wheel (m) [A3]	7.66	3.94
Dist. from CG to Axle (m) [A4]	4.64	2.46
Half Lateral Distance Between Centres of Tire Contact on Trailer Axles (m) [TRA3]	1.03	1.03
Trailer Tandem Axle Load Transfer Coef. [GAM4]	0.15	0.15

*Derived from reference (11)

**Payload = 13607 kg (30000 lb)

Table 5 Tire Data*

DESCRIPTION	VALUE
Cornering Stiffness per tire (N/deg) [CALF]	4300
Peak Tire-Road Coef. [MUP]	0.47
Sliding Tire-Road Coef. [MUS]	0.25
Slip corresponding to Peak Friction Coef. [SP]	0.15

Note: The same tire characteristics were modelled on all axles.

*Derived from references (11) and (12)

The four combinations studied were referred to as follows:

Combination 1: 5.6 m Tractor / 14.6 m Trailer
Combination 2: 5.6 m Tractor / 8.2 m Trailer
Combination 3: 3.6 m Tractor / 14.6 m Trailer
Combination 4: 3.6 m Tractor / 8.2 m Trailer

Many of the values chosen were exaggerated to challenge the yaw stability of the vehicle and to promote its jackknife tendencies. For example, the height of the loaded trailer's CG above the ground was very low. The lower CG values reduced the roll-over tendencies of the vehicle, which was a stopping criterion for the program, and promoted the occurrence of jackknife.

The spread between MUP (0.47) and MUS (0.25) (Table 5) may seem large, but this again promoted the occurrence of a jackknife. When the brakes were locked to induce jackknife or trailer swing, the shear forces at the tire-road interface rapidly changed from a maximum value (due to MUP) to a minimum value (due to MUS).

5.3.2 Simulated Manoeuvres

Two manoeuvres, a constant radius turn and a lane change, were selected for analysis. Each manoeuvre was successfully completed at the specified speeds, then jackknife and trailer swing were induced through aggressive brake application. These two simulation runs were combined to generate a "performance range" or envelope against which the simulated AJS runs could be measured.

Constant Turn Radius

The constant radius turn simulation modelled two curves with different radii. A 90 m radius turn with a MTO design speed of 50 km/hr was modelled at speeds of 30 km/hr, 50 km/hr, and 57.5 km/hr. These speeds equate to theoretical lateral accelerations of 0.08g, 0.22g, and 0.30g respectively. The lateral acceleration of 0.30g was chosen as a maximum limit of the program because values higher than this tended to produce questionable results. This complied with El-Gindy's findings (13) that the results from simulated lateral accelerations of greater magnitude progressively worsen with respect to measured data. Similarly a 250 m radius turn, with a MTO design speed of 80 km/hr, was simulated at the 0.30g limit (97.5 km/hr).

Lane Change

Analysis of tractor semitrailers performing a lane change manoeuvre was incorporated into the study for a number of reasons. Braking combined with a lane change represented a possible evasive manoeuvre. For example, braking before a lane change represented a possible panic situation where, to avoid collision, the driver braked heavily and then attempted to steer into an adjacent lane or represented a situation where a driver initiated a lane change and found it necessary to brake hard during the manoeuvre. The

transient responses of a tractor semitrailer manoeuvring a lane change may display similar angular characteristics to that of jackknife and trailer swing. These characteristics would have to be known to accurately distinguish one from the other.

Lane change simulation results should be studied with caution as these results are very open to interpretation. For example, a particular configuration may perform very well in one lane change manoeuvre and very poorly in another. For this reason a common basis for comparison had to be chosen. The maximum attainable steer rate of an average driver, as given by M. El-Gindy, was chosen for this common basis. Numerically this equated to a sinusoidal input of amplitude + 5° to - 5° over a time span of 2.2 seconds. Based on this, all lane change simulations were given an initial velocity of 80 km/hr and the maximum steer rate as inputs.

Due to the variances in steer angle, braking at various points in a lane change produced dramatically different results. For this reason, brake applications at four different points in time were modelled.

5.3.3 Methods of Inducing Jackknife and Trailer Swing

As previously discussed, a jackknife occurs because of the saturation (or limitation) of lateral forces at the tire road interface. This loss of force may be due to many influences, such as aggressive steer inputs, large side wind loads, high torques at the wheels, etc. However, the most common cause of a jackknife is excessive braking, usually to the point of tire lockup. It should be noted at this point that the onset of jackknifing corresponds with partial skidding rather than with locked wheels (14).

In order to maintain consistency, a very aggressive brake force was induced at the appropriate wheels when the lateral side force of a particular tire was required to be saturated (lock the wheel). To be specific, all wheels on the tractor tandem axles were locked in this fashion to induce a tractor jackknife, and similarly all tires on the trailer tandems were locked to induce trailer swing.

An aggressive brake force was a braking force, large enough to lock the wheels under all circumstances, applied over a short period of time. This method of application was required due to the variances in shear forces at the tire road interface which result from load transfers at different points in time. Thus, lock-up was assured to occur quickly for all manoeuvres and all configurations in all instances. The range and magnitudes associated with this aggressive brake application were; from 0.0 N. to 50,000 N. over 0.5 seconds (refer to Appendix "D" for brake input to the program).

5.4 Characteristics of Jackknife and Trailer Swing

Before attempting to alleviate jackknife and trailer swing, a close investigation into the characteristics of these forms of instability was carried out. The approach taken was to first simulate "successful" manoeuvres by negotiating the intended path in the absence of braking, then,

to induce either jackknife or trailer swing through the braking methods discussed in Section 5.3.3.

The variances in the simulation variables were studied in an attempt to distinguish and classify the characteristics of jackknife and trailer swing. Figure 7 illustrates how the articulation angle of a tractor semitrailer combination negotiating a constant radius turn varied from steady state in both a jackknife and a trailer swing situation.

The most obvious observation made from the diagram was that jackknife and trailer swing behave in dissimilar manners, even though they both are products of the loss of lateral force at the tire-road interface.

5.4.1 Characteristics of Jackknife

The loss of lateral forces at the tire road interface is the root of the problem. Figure 8 shows the lateral forces on the tires of the tractor tandem axles in a jackknife situation. At time t=0 the vehicle was in line, zero articulation, and travelling at a speed of 50 km/hr. The steering angle was ramped from 0° at t=0 s to 4.24° at t=0.5 s and the rear tractor wheels (3, 4, 5, 6; refer to Figure D6 in Appendix "D" for numbering convention) were locked by applying a brake force of 0 N at t=2.0 s to 50,000 N at t=2.5 s.

The lateral forces increased from 0 N to a steady state limit from time t=0 s to the point just before brake application, state point # 1. This can be explained by realizing that the vehicle was "in-line" at t=0 and travelling forward at a constant velocity of 50 km/hr, therefore no lateral forces existed since the slip angle remained at zero (recall Figure 8). The magnitudes of the lateral forces on the tandem axles of the tractor (5-6 and 3-4) differed because of the "aligning moment" that existed during the turning manoeuvre.

The variances in the lateral forces between the tires on the same axle (i.e. 3 and 4 or 5 and 6) show the dynamic conditions of cornering. A load transfer existed due to the height of the tractor CG and the impending lateral acceleration (quasi-static computations). Thus, the inside wheels, 4 and 6, carried a smaller portion of the load during cornering than did the outside wheels, 3 and 5. When a brake force was applied, the inside wheels, 4 and 6, locked prior to 3 and 5, due to the decreased normal load. Between state points #1 and #2 (Figure 8) the lateral forces were reduced to a minimum.

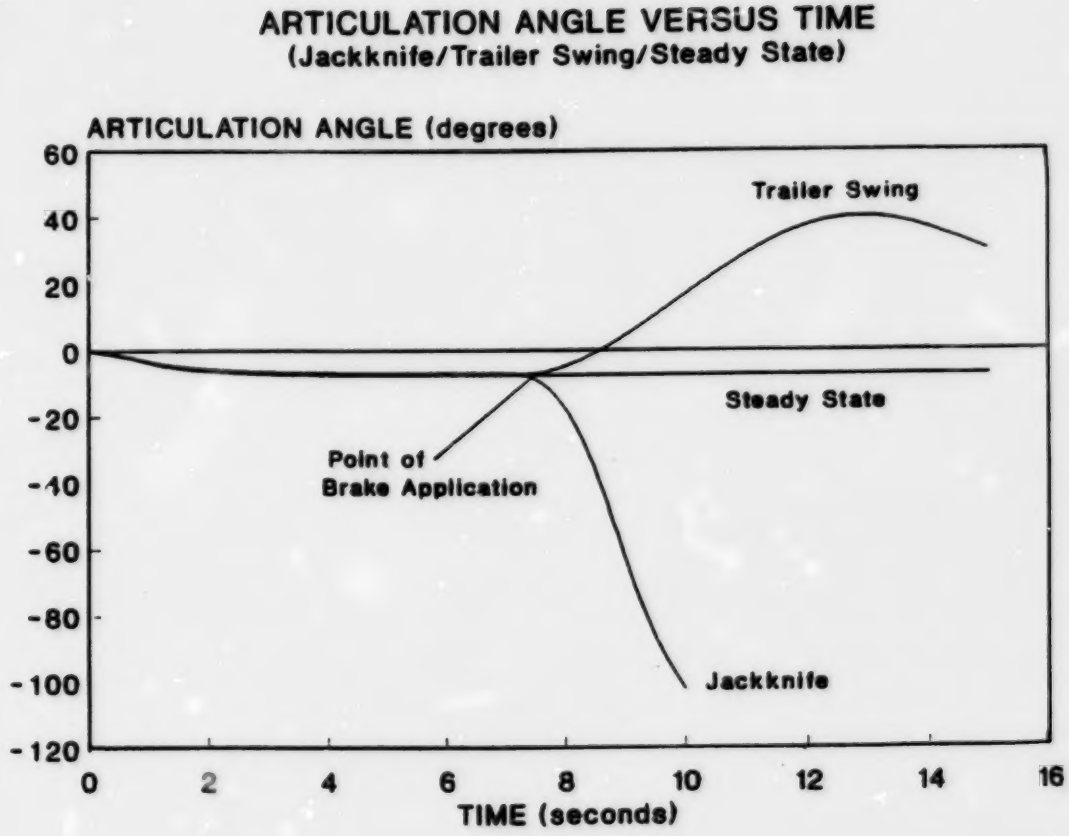


Figure 7 Typical Characteristics of Jackknife and Trailer Swing During A Constant Radius Turn

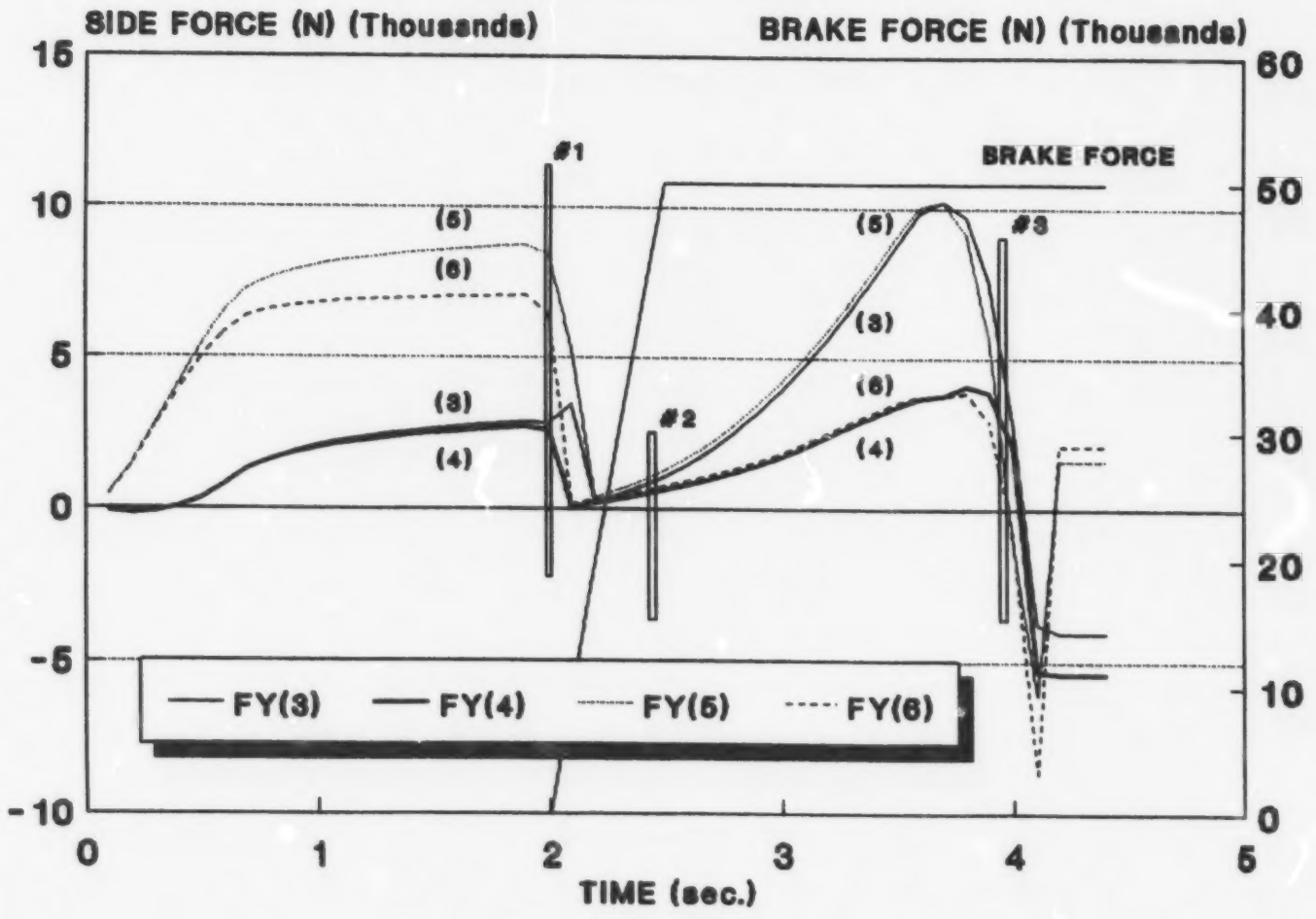


Figure 8 Lateral Forces on Rear Tractor Tires During Jackknife

The reduction in lateral forces, in this study, were due directly to braking. Initially, up to state point #1, all tires were free rolling and all shear forces at the tire-road interface acted laterally on the tires. The brake forces, FS_x, increased with little effect on the lateral forces until a certain value was reached. When this value was exceeded a "lock-up" condition existed and the resultant frictional force changed to directly oppose the direction of motion, which at this point was mainly along the x-axis of the tire.

These shear force characteristics were explained by the "friction circle" model described by Warner et. al. (15). The model stated that the total available friction force was given by $F_{max} = \mu F_z$ and it acted in the direction opposite to the relative motion of the tire and road surface. If the vector sum of the longitudinal and lateral forces did not exceed F_{max} the tire continued to track and the wheel rotated. When lock-up occurred rotation ceased and the friction force swung around the "circle" acting in a direction which opposed relative motion. Using this friction circle concept, the maximum available side force, F_y, was given by:

$$F_y = (\mu^2 F_z^2 - F_x^2)^{\frac{1}{2}} \qquad [1]$$

Figure 9 schematically represents the model. Actual tires exhibit characteristics that are more accurately modelled as a "friction ellipse" having major and minor half-axes equal to $\mu_x F_z$ and $\mu_y F_z$, respectively. At small slip angles the longitudinal shear force (the major axis) is much greater than the lateral forces, hence the friction ellipse concept holds. At larger slip angles the two forces approach equality and are better represented by the "friction circle" model.

Jackknife occurred when the lateral forces on the outside tires, 3 and 5, reach saturation (between state points #1 and #2 of Figure 8). Beyond state point #2 the lateral forces on the tires increased as the orientation of the tractor departed from the X-direction of motion (as it rotated out of control about the king pin). Tires 3 and 5, the outside tires, exhibited lateral forces of greater magnitude than tires 4 and 6, the inside tires, because of load transfer characteristics previously discussed.

The purpose of this study was to investigate the feasibility of anti-jackknife devices about the fifth wheel. Implementation of such a device involved studying more than just the shear forces at the tire-road interface. Characteristics about the fifth wheel, for this reason, were very important in the study. To aid in the discussion, Figure 16(b) (presented in Section 6.5.2) visually represents the orientation and position of a tractor-semitrailer configuration involved in a brake induced jackknife while negotiating a constant radius turn. Jackknife, although caused by loss of lateral forces at the tire-road interface, is characterized by its articulation attributes. As shown in Figure 7, the articulation angle in jackknife rapidly diverged from its steady-state value. The rate of divergence can be as high as 65 deg/sec for the configurations studied. If 15° of articulation is used as a benchmark indicator for jackknife, it is quickly reached in as little as 0.5 seconds.

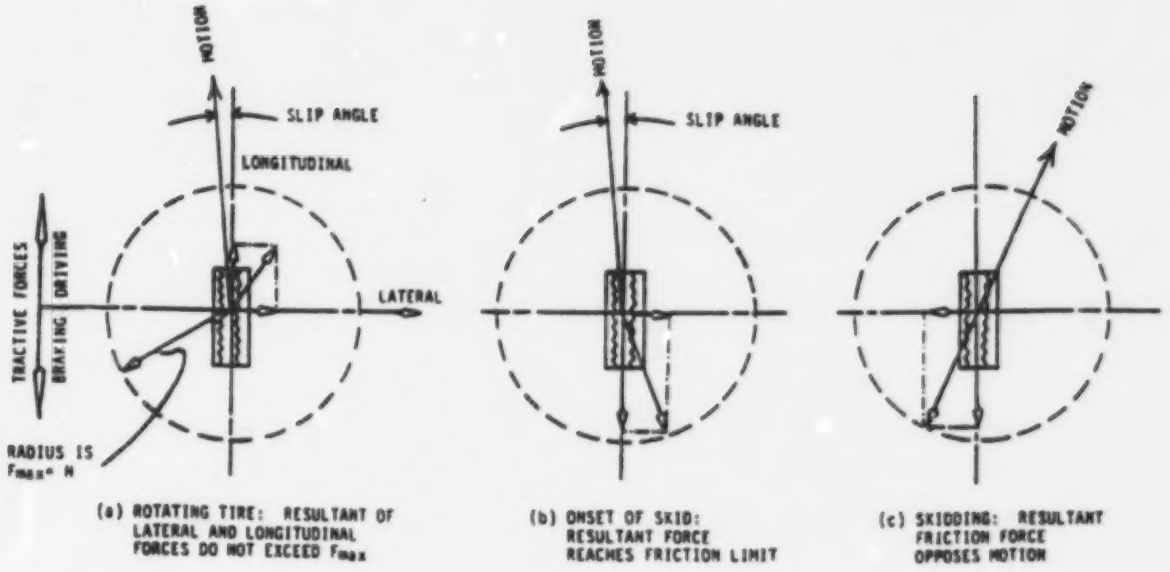


Figure 9 The Friction Circle Model (15)

An increase in forward velocity will increase the severity of the jackknife. This is due to the dependency of the lateral acceleration, in cornering, on the velocity of the vehicle ($a_y = V^2/R$). Thus, the centrifugal ($F_c = m \cdot a_y$) and inertial forces acting on the vehicle play a more significant role. The lateral forces acting on the tires can produce only a limited amount of resistance. A larger force that aggravates the vehicle jackknife tendencies results when this resistance is overcome by externally acting forces.

From the material presented in Appendix "E", it is evident that the characteristics of a jackknife are very dependent upon the vehicle configuration and payload. A shorter and lighter tractor will experience a more aggressive (or rapid) jackknife than a longer heavier one. This is due to the lower inertial resistance, lower normal load (hence lower friction forces), and the smaller effective rotational moment arm (wheelbase). An increased trailer payload will increase the severity of a jackknife because the forward momentum of the trailer produces a larger force on the same moment arm rotating the tractor about its front axle.

Investigation into the occurrence of jackknife without braking was carried out by simulating a tractor semitrailer subjected to a very aggressive steer input. The simulation proved that jackknife can occur without brake application when the induced lateral forces on the vehicle exceed the lateral restraining forces at the tires. The resulting jackknife showed less severe articulation angles and rates than the brake induced jackknife. In this instance all frictional forces that oppose motion act laterally on the tires and are not reduced by the longitudinal forces of braking. Recall that there is a finite limit to the total available friction force which is a vector sum of the lateral and longitudinal shear forces at the tire (friction circle model).

5.4.2 Characteristics of Trailer Swing

A visualization of trailer swing is depicted in Figure 18(b) (presented in Section 6.5.2). As with jackknife, trailer swing occurs because of the loss of lateral side forces at the semitrailer tandem tires. This loss of side force is shown in Figure 10. The semitrailer tires exhibit identical characteristics to those of the rear tractor tires. Lateral forces on all tires begin at zero and build to a steady state value. Tires 9 and 10 experience larger lateral forces than 7 and 8 because of the aligning moment that exists. In trailer swing, similar to the tractor rear tires in jackknife, wheels 8 and 10 experience lock-up before the outside tires, 7 and 9, because of the lateral load transfer that takes place during cornering. As the semitrailer swings away from its steady-state articulation angle the lateral forces increase, thereby resisting the motion.

Evident from Figure 7, trailer swing does not occur as rapidly as jackknife. It reaches maximum values in the order of only 19 deg/sec for the configurations modelled. In fact, trailer swing tends to regain stability without any form of compensation, as may be seen from the reduction in the articulation angle as time elapses (Figure 7). Another difference between trailer swing and jackknife is that the directional change in articulation occurs in opposite directions.

LATERAL FORCES ON TRAILER REAR TIRES DURING TRAILER SWING

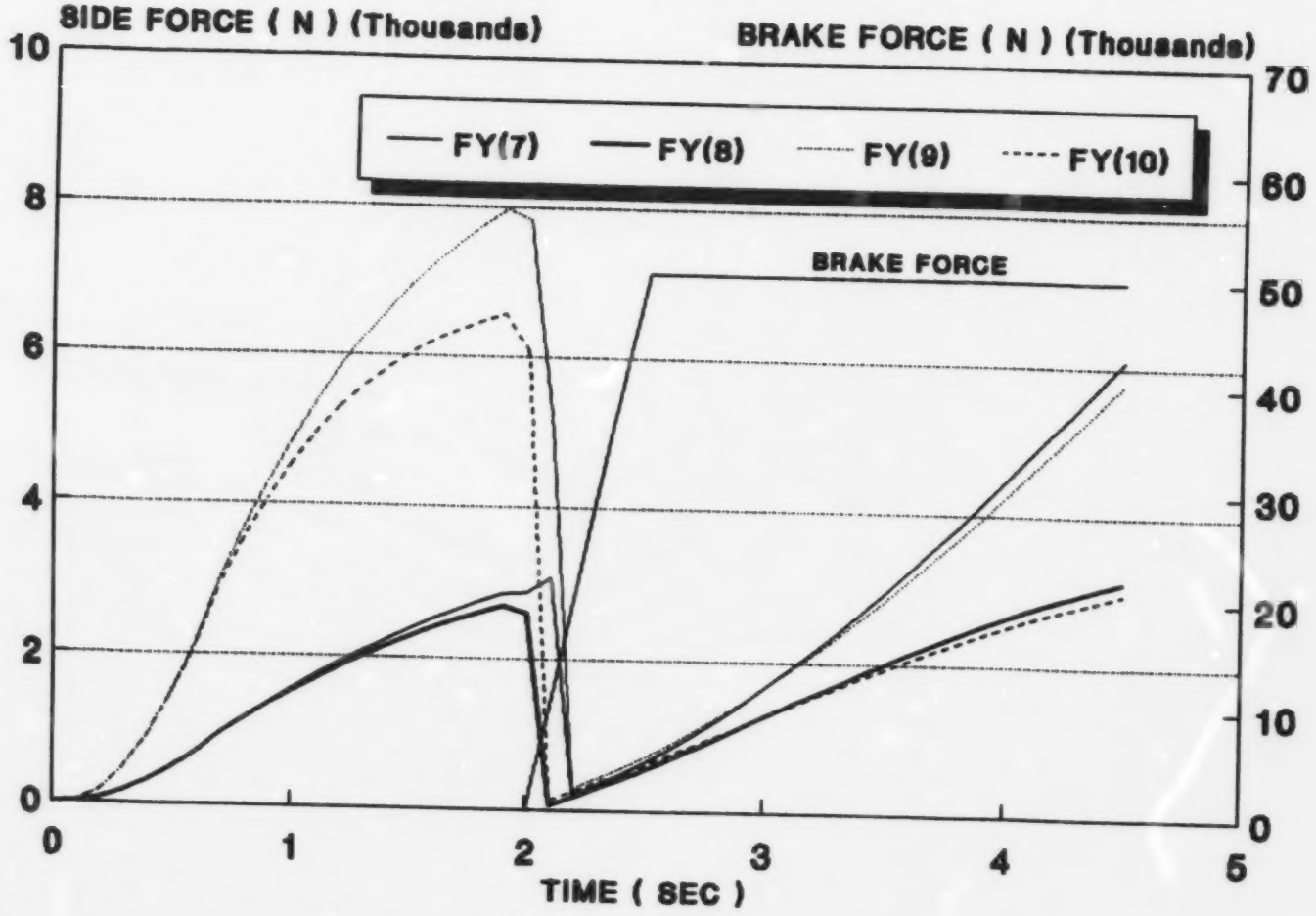


Figure 10 Lateral orces on Semitrailer Tires During Trailer Swing

Jackknife produces a negative articulation while trailer swing articulates positively. Figures 16(B), and 18(B) presented in Section 6.5.2 will aid in visualizing these occurrences.

Using 15° articulation as an indicator for trailer swing does not have the same implications as it does for jackknife. Because trailer swing is a milder form of instability, the 15° mark is typically reached about two seconds from the onset of tire lock-up, as opposed to 0.5 s for jackknife. Also, fifteen degrees of articulation in trailer swing does not indicate a point beyond which there is no control, as it implies with jackknife.

The above control difference was demonstrated in two instances. A standard simulation run (Figure 7) showed that even after 41° of articulation the trailer tended to stabilize on its own while the brakes remained locked. A second test series involved two simulation runs, one which induced jackknife, and another which induced trailer swing (both through wheel lock-up). These simulations differed from previous runs in which the brakes remained locked. In this series the brakes were released when approximately 15° of articulation was reached. The results showed that the jackknife induced run continued to jackknife, although slightly less rapidly, after release of the brakes. The trailer swing induced run quickly regained stability and approached steady-state articulation upon release of the brakes.

Torques about the fifth wheel resulting from trailer swing are much larger than the torques produced in a jackknife. The greater mass of the trailer over the tractor, the corresponding increase in inertia, and the larger moment arm from the fifth wheel to the trailer CG contribute to this greater torque. Two factors that directly influence the torque produced at the fifth wheel are the trailer payload and the speed at which the vehicle is travelling. An increase in either one will produce a corresponding increase in the moment about the fifth wheel.

As in the case of jackknife, a shorter trailer will experience a more rapid trailer swing than a larger trailer with the same payload. This is due to its decreased inertial resistance. A similar conclusion may be made for an empty/loaded trailer comparison.

The literature and the statistics tend to indicate that trailer swing contributes to fewer accidents than jackknife. This is a hard fact to prove as police officers have no way of knowing whether jackknife or trailer swing were the cause or result of the accident.

5.5 Sensing of Jackknife and Trailer Swing

Sensing plays an important role in the operation of active anti-jackknife devices (devices which restrain jackknifing after its occurrence). The answers to the following questions must be known in order to effectively limit the jackknife or trailer swing action:

- Is jackknife or trailer swing happening?
- If so, is it jackknife or trailer swing?
- Should a restraining torque be applied?

- If so, how much?
- If a restraining torque is applied does it affect vehicle stability or controllability?
- When should the device disengage?

The large variances in configurations, manoeuvres, and payloads make the efforts involved in assigning an absolute number to a variable (or combination of variables) very difficult. For example, the rate of articulation of one configuration experiencing a jackknife (or trailer swing) in a particular manoeuvre (say a constant radius turn) may not significantly differ from another configuration involved in a more dynamic manoeuvre such as a lane change. The problem therefore lies in the transient responses of the vehicles.

Any deviation from the steady state value of a particular variable is easily detected. For example, if the articulation rate significantly deviates from zero, an adverse condition may be said to exist. Similarly, the same conclusion may be drawn if the articulation angle or the yaw rate of the tractor deviates from its steady-state constant value. Even if it was possible to ignore transient responses, values associated with steady-state variables differ with each situation. This variance is due to the non linear characteristics of the vehicle. For example, a typical vehicle may change from understeer to oversteer as the lateral acceleration increases.

A means of controlling jackknife had successfully been implemented in situations where the vehicle parameters remain relatively constant, such as in articulated buses. An anti-jackknife system for buses developed by the Hungarian Automotive R & D Establishment (16) uses optical sensors for determining steering and pivot angles. These sensors feed data to an electronic control unit which compares the data with programmed values for a specific turn radii. With any angular deviation from the programmed values, valves in a fluid circuit interconnecting hydraulic cylinders are closed to prevent further movement between vehicle sections. A device such as this has its shortcomings when applied to tractor semitrailers. The "Look-up-table" of the electronic control unit applies only to a single combination. This defeats one of the major attributes of the tractor semitrailer: interchangeability. Associated with interchangeability is the corresponding variance in vehicle characteristics which complicates the problem of sensing.

As was discussed in sections 5.4.1 and 5.4.2, both jackknife and trailer swing are events that result from saturation or limitation of the lateral forces at the tire-road interface. Bourassa et. al. (6) quotes Susemihl by stating that an approximate side force may be derived from measurements of translational tractor lateral acceleration, angular tractor acceleration and the lateral force of the trailer in the fifth wheel. As stated by Susemihl, two accelerometers and a strain gauge may be used to physically evaluate the side force.

The discussion to follow presents the investigation of the characteristics of the simulation variables and how these characteristics may be used as an indicator of jackknife and/or trailer swing.

5.5.1 Observations from Simulation

Table 6 contains a list of some of the simulation variables and how they may or may not be used to indicate (or sense) jackknife and/or trailer swing. The table was constructed with the intent of finding the value of a single variable or a simple combination of variables that would, under any condition, indicate jackknife or trailer swing.

In the scope of this study, an absolute indicator for both jackknife and trailer swing, aside from the generally accepted 15° of articulation, was not found. However, a few possible indicators of tractor jackknife appeared evident.

For example, an articulation rate of ± 15 deg/sec did indicate the occurrence of a jackknife. It can not be confused with the transient responses of normal manoeuvres, as simulations indicated that transient responses exhibited lower rates of articulation. The draw back to this indicator was that when the articulation rate reaches a value of ± 15 deg/sec, the vehicle has already been articulated past 15 degrees. It therefore may be termed a late indicator.

The side slip angle of the tractor appeared to be an accurate indicator for jackknife. One drawback was that when ± 4 degrees of side slip was achieved the vehicle had articulated to approximately 12°. This may correspond to as much as 7° of additional articulation from the onset of jackknife. At 12°, impending forces are relatively high (refer to Section 6.3) and a devices effectiveness may be reduced. Another limitation is the required sensing, the longitudinal and lateral velocities must be accurately measured as side slip is equal to $\tan^{-1}$ (V-vel/U-vel). It is very difficult to get reliable measurements of slip angle, even in a research environment.

A better indicator of jackknife was the tractor yaw rate which was determined with respect to the reference heading at time t=0. When a value of ± 15 deg/sec was reached, a jackknife was in its early stages. Similarly, a semitrailer yaw rate greater than or equal to ± 15 deg/s indicated trailer swing. This semitrailer yaw rate was not available as output from the program, but it could be calculated knowing the tractor yaw rate and vehicle articulation rate ($\dot{\Gamma}$ = Semitrailer yaw rate - $\dot{\Psi}$). Actual measurement of these values poses a problem as there is no fixed reference from which to measure. As a result, complex measuring devices would be required.

A more detailed investigation of the sensing problem is required. Close examination of the vehicle articulation rate may produce a value that can be used to indicate jackknife at an earlier stage. Investigations should also be conducted to determine if there is a combination of variables that will predict the occurrence of a jackknife.

Table 6 Evaluation of Output Variables to Determine a Basis for Sensing Jackknife and Trailer Swing

VARIABLE	JACKKNIFE	TRAILER SWING	COMMENTS
Articulation Angle (Γ)	> ± 15°	> ± 15°	- Very late indicator for jackknife, some standard manoeuvres may easily exceed this value.
Articulation Rate ($\dot{\Gamma}$)	> ± 15 deg/sec	No	- The lane change manoeuvres reached values just below this. - Late indicator, articulation has already exceeded 15°. - Trailer swing rates are generally lower and do not apply.
Articulation Acceleration ($\ddot{\Gamma}$)	No	No	- Articulation acceleration in transient responses of normal manoeuvres may be greater than what is experienced in jackknife and trailer swing situations.
Tractor Yaw Angle (Ψ)	No	No	- This variable is referenced to the vehicle heading at time t=0. - Constantly changing unless travelling in a straight line.
Tractor Yaw Rate ($\dot{\Psi}$)	> ± 15 deg/sec	No	- This is an early indicator of jackknife that does not occur during the transient response of normal manoeuvres. - Does not apply to trailer swing.
Tractor Yaw Acceleration ($\ddot{\Psi}$)	No	No	- Lane changes manoeuvres may be as high as 43 deg/sec2, - Jackknife can occur with angular accelerations less than this.
Side slip angle of tractor	± 4 deg	No	- This does not apply to trailer swing because the tractor remains relatively stable. - ± 4 degrees are reached before 15° of articulation in a jackknife situation. - Good indicator, but not as good as Tractor Yaw Rate ($\dot{\Psi}$).
Semitrailer Yaw Rate	No	> ± 15 deg/sec	- An early indicator for trailer swing. - The value is above all normal transient values.

6.0 AJS DESIGN SYNTHESIS

The design of an active anti-jackknifing system can be separated into two tasks. One task is the development of a system for sensing the onset of jackknife or swing and the other is the development of a device to control the articulation of the vehicle during jackknife or trailer swing.

The sensing portion of the AJS design was dealt with under the previous section. The development of the articulation control device and the performance results obtained when this design was tested in computer simulations of jackknife and trailer swing are discussed below and summarized in the Flow Chart, Figure 11.

6.1 Methodology

The following steps were performed:

1) Establish the performance criteria of an ideal AJS.

2) Conduct a simplified analysis of a tractor semitrailer during jackknife and trailer swing to determine the magnitude of the resistive moments that an anti-jackknifing device would have to generate and to establish the stability of the unit when the AJS is activated.

3) Determine, classify and appraise all the different generic methods of controlling jackknife or trailer swing established from the patent search of AJS devices as well as unpatented new ideas developed by the research team.

4) Investigate the design feasibility, specifically appraising the upper limit of resistive torque that could be realistically achieved at the fifth wheel by the particular AJS method.

5) Realistically model and evaluate these methods by computer simulations of jackknife and trailer swing.

6) Determine the impact driver reaction would have on the simulation results in jackknife and trailer swing situations.

7) Examine more closely the devices which have shown merit to determine the effect that those devices might have on the normal and safe handling characteristics of the vehicle.

8) Evaluate the effect of front axle brakes and of ABS on the performance of the AJS.

With respect to computer simulation, the method of brake application is very important. As was discussed in section 5.3.3, jackknife and trailer swing were induced through an aggressive brake application on specified axles. To reiterate, this type of aggressive brake application is very uncommon in real life occurrences and makes the anti-jackknife requirements in this study very stringent.

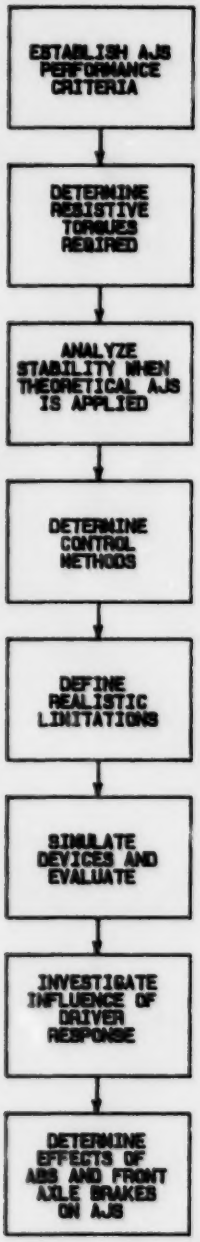

ESTABLISH AJS PERFORMANCE CRITERIA
DETERMINE RESISTIVE TORQUES REQUIRED
ANALYZE STABILITY WHEN THEORETICAL AJS IS APPLIED
DETERMINE CONTROL METHODS
DEFINE REALISTIC LIMITATIONS
SIMULATE DEVICES AND EVALUATE
INVESTIGATE INFLUENCE OF DRIVER RESPONSE
DETERMINE EFFECTS OF ABS AND FRONT AXLE BRAKES ON AJS

Figure 11 Design Synthesis Methodology

6.2 Performance Criteria

It was necessary to establish performance criteria at the onset of the design phase in order to provide a basis upon which to evaluate the various forms of jackknife control. As well, these performance criteria would guide the development of new control methods.

The criteria that were formulated are:

- the AJS must prevent the articulation angle, during tractor jackknife, from exceeding a range of ± 15° (from the inline position) in a time span of 1.5 seconds from the onset of the jackknife. The smaller the change in articulation angle due to tractor jackknife or trailer swing, in the 1.5 second time span, the better the device was rated. There are two important qualifiers to this criteria. Firstly, the restriction of the articulation angle must not result in a loss of lateral stability of the entire vehicle. Secondly, the restriction of the articulation angle must not make steering changes impossible during AJS activation.

This criterion was based on the commonly accepted thought that if a vehicle has articulated beyond a range of ± 15° due to jackknifing the vehicle has reached a point where regain of control may no longer be possible. Corrective action to regain control may include steering changes, switching from a constant brake application to a pumping action for the brake induced jackknife or trailer swing, or depressing the clutch or releasing the accelerator for the power jackknife. The 1.5 seconds is a conservative driver reaction time based on a slower night-time reaction (17).

- the AJS must prevent the tractor and trailer from intruding more than one metre into a neighbouring lane during the 1.5 second time span from the onset of jackknife or trailer swing.

This criterion was adapted from the authors of the "Performance Evaluation of Several Jackknife Control Devices" (5). They concluded that one metre was the maximum intrusion allowable without precipitating an accident. After the 1.5 second time span, the evaluation was more subjective - based on how it was thought the driver might react when using a particular AJS.

6.3 Empirical Model and Analysis

To assist in the understanding of jackknife and trailer swing phenomena, a simplified analysis was conducted early in the design phase. This analysis was meant to determine the relative magnitudes and directions of the forces acting on the vehicle during jackknife and trailer swing and to determine the AJS resistive torque required to prevent this occurrence. As well, the lateral stability of the vehicle during jackknife and trailer swing was of critical interest when the AJS control methods are applied.

The following discussion outlines how this analysis was carried out, explains the results that were obtained and discusses the assumptions that were used to attain these results.

6.3.1 Model of Tractor-Trailer During Jackknife and Trailer swing

Models or free body diagrams (F.B.D.) of a tractor semitrailer during brake induced jackknife and trailer swing while the vehicle was negotiating a corner were constructed. Figures 12 and 13 depict all the forces acting on the vehicle during this manoeuvre. A brake induced (wheels locked) jackknife and trailer swing was chosen as braking is the most frequent cause of jackknife or trailer swing. However, the results obtained would be representative of a power jackknife, as the same phenomenon of loss of tire lateral forces occurs with it as well.

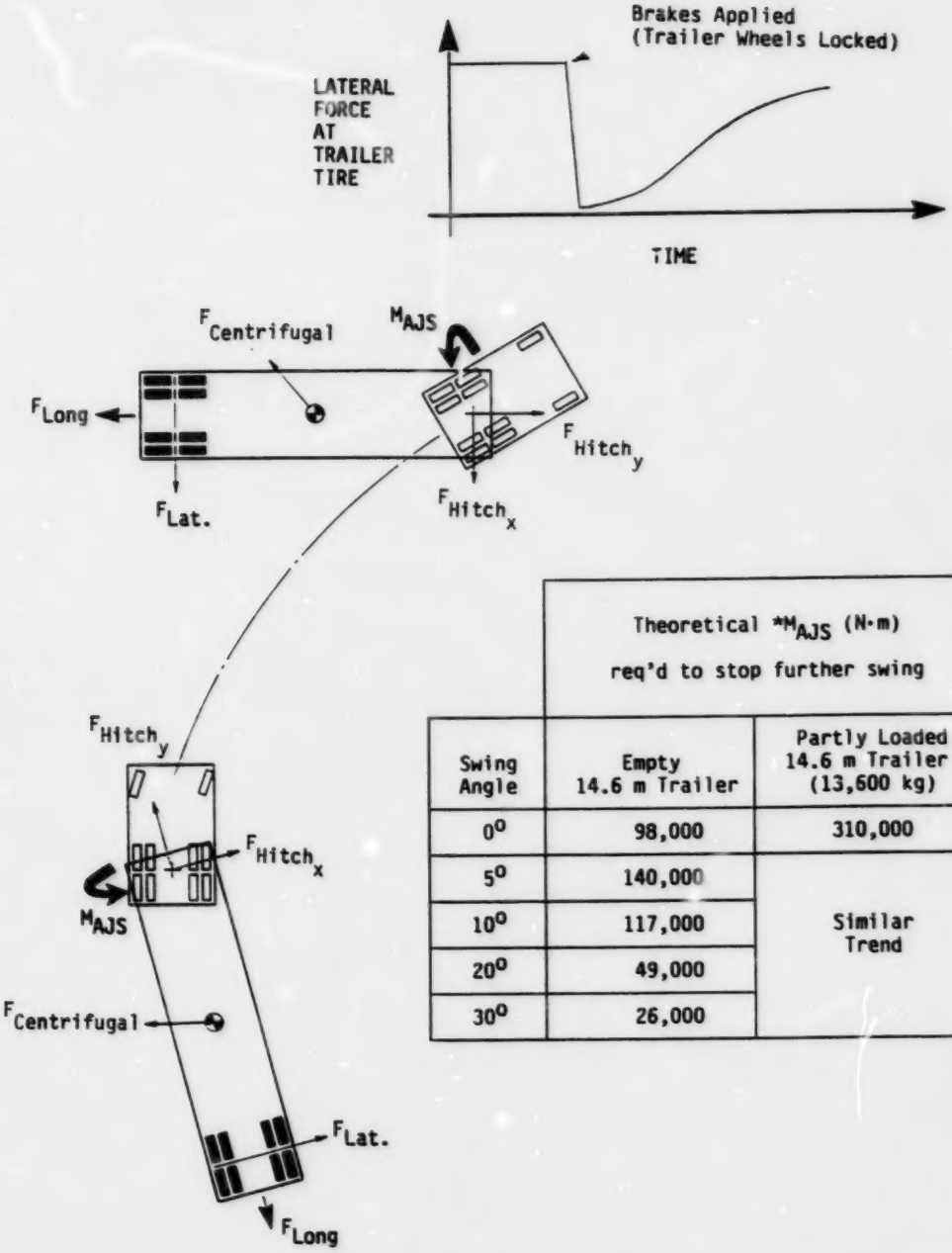
On examining the F.B.D. for trailer swing (Figure 12), it is evident that if the tire lateral forces are sufficiently reduced (due to wheel lockup, as seen in Figure 10) the trailer will swing outward, under the action of centrifugal force, as the vehicle travels forward. It is the tires which create the centripetal force (directed toward the centre of curvature) necessary for cornering - represented as centrifugal force (opposite in direction to the centripetal force on the F.B.D. This means any AJS device must supply sufficient torque to balance the loss of tire lateral forces if trailer swing is to be prevented.

Examining the F.B.D. for tractor jackknife, Figure 13, it is again evident that it is the influence of centrifugal force which initially causes the vehicle to jackknife. However, as the jackknife progresses the tractor will have a tendency to pivot about the front tires, thereby adding to the resistive moment that the AJS must supply. This increase in resistive moment is due to the increasing moment arm as the tractor pivots. If the front tires could be pointed in the direction of tractor travel and the front brakes released, this moment would be reduced.

6.3.2 Analysis of AJS Torque Requirements

The resistive torque that the AJS must be able to supply was calculated. As shown by the F.B.D.'s, this torque will be largely a function of the centrifugal force, the normal load, and the coefficient of friction at the tire/road interface. In order to determine a realistic operating range (torque requirements) for the AJS, vehicle configurations and cornering speeds were chosen so as to represent typical operating conditions.

The calculated resistive torques are summarized in the charts on Figures 12 and 13. They represent the torque that would be required to lock the articulation from the very onset of jackknife or trailer swing through the various degrees of jackknife or trailer swing.



Swing Angle	Theoretical *M_{AJS} (N·m) req'd to stop further swing: Empty 14.6 m Trailer	Theoretical *M_{AJS} (N·m) req'd to stop further swing: Partly Loaded 14.6 m Trailer (13,600 kg)
0^{o}	98,000	310,000
5^{o}	140,000	Similar Trend
10^{o}	117,000	
20^{o}	49,000	
30^{o}	26,000	

*Calculated Moments are based on Combination 1, R = 90m (295 ft), Speed = 50 kph (30 mph)

Figure 12 F.B.D. of Trailer Before and After Brake Induced Trailer Swing in a Constant Radius Turn

Examining the AJS torque values for trailer swing, it can be seen that the resistive torque increases in the first 5° to 10° and then gradually decreases towards zero as the trailer approaches a trailer swing angle of 40 to 50° from the original steady-state articulation angle. Initially the trailer gains rotational inertia and therefore requires more torque to stop. As it swings further, the reduction in vehicle speed due to braking decreases the centrifugal force and increases the tire lateral forces resulting in a decreasing resistive torque being required. The tire lateral forces increase as the semitrailer swings because the friction force acts increasingly closer to the lateral direction on the tire (Recall the friction circle model in Section 5.4.1).

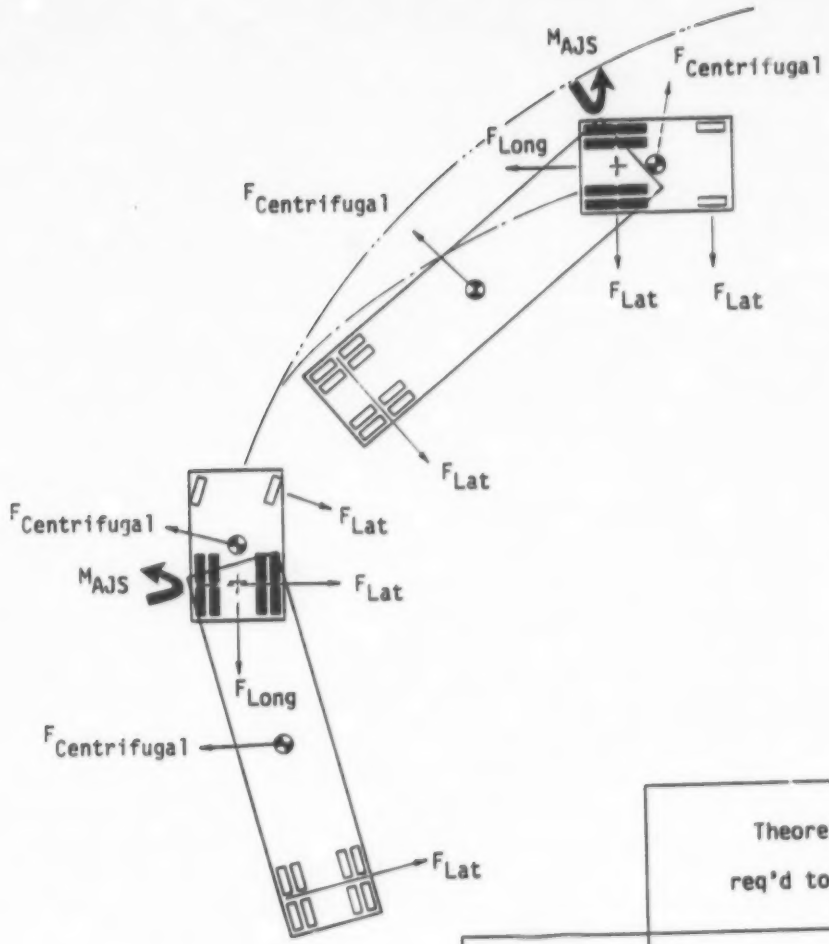
Examining the AJS torque values, in terms of the effect of payload, it can be seen that adding payload has a dramatic effect on increasing the level of torque required from the AJS.

The AJS torque values required to prevent tractor jackknife at the very onset of jackknife is about half that for trailer swing. As well, increasing the trailer payload does not increase the required resistive torque as dramatically. It is interesting to note, however, that the torque required to prevent jackknife does not decrease with increasing angle as rapidly as it does for trailer swing.

The vehicle configuration and speed were chosen to represent typical operating conditions that would promote the occurrence of swing or jackknife. This was accomplished by using the longest wheelbase trailer and tractor and by using the maximum design speed for a specific curve radius. For Ontario highways the maximum design speed for cornering typically equates to a lateral acceleration of about 0.22 g. It was determined that increasing the speed to attain a threshold lateral acceleration of about 0.3 g results in a 37% increase in the torque required from the AJS for a 16% increase in speed. This is in line with expected results as centrifugal force is proportional to velocity squared. A lateral acceleration of 0.3g is typically used as a cornering threshold for tractor-trailers.

The calculations for the torque required to prevent trailer swing at the very onset were quite straightforward. However, once the trailer had gained rotational inertia the analysis was more difficult. Assumptions were made as to the time span or swept angle that the AJS would require to reduce the rotational inertia to zero (consult Appendix "F" for details on these assumptions). As a result, the calculated AJS torque value at the onset of trailer swing was quite accurate, but the torque values at various stages into trailer swing were more of an indication of the amount of torque that was required.

The analysis of tractor jackknife was even more complex, especially once the tractor angular acceleration had increased. As a result, the calculated resistive torques were indicators rather than exact values of the resistance required to inhibit the occurrence of jackknife (consult Appendix "F" for details of the assumptions made).



Jackknife Angle	Theoretical M_{AJS} (N·m) req'd to stop further swing	
	Empty 14.6 m Trailer	Partly Loaded 14.6 m Trailer (13,600 kg)
0^{o}	52,000	92,600
5^{o}	80,000	Similar Trend
10^{o}	100,000	
20^{o}	95,000	
30^{o}	95,000	

*Calculated Moments are based on Combination 1, R = 90 m (295 ft), Speed = 50 kph (30 mph)

Figure 13 F.B.D. of Tractor and Trailer Before and After Brake Induced Tractor Jackknife in a Constant Radius Turn

Other details affecting the accuracy of the results are as follows. The tire lateral forces were obtained from the computer simulation for the particular configuration and manoeuvre, thereby improving the accuracy of the results. Similar to the computer simulation no braking was applied to any wheels other than on the locked axles.

6.4 Control Methods

In order to determine the most suitable AJS control method, it was necessary to establish the generic methods used to control jackknife and trailer swing. This section outlines these methods and presents the advantages and disadvantages of each method. As well, it details our investigation of the design feasibility of these control methods.

6.4.1 Classification and Evaluation of Generic AJS Control Methods

As shown in Figure 14 there are two basic ways of approaching the jackknife control problem. One method is to control the articulation, usually by some means about the fifth wheel, and the other is to prevent wheel lock-up which is the major cause of jackknife and trailer swing. Prevention of wheel lock-up can be achieved by either of two methods: anti-lock brakes or load sensing with brake proportioning. Load sensing with brake proportioning has no ability to prevent a power jackknife or any other non-brake induced types of jackknife. However, new ABS technology is showing promise in methods of spin control that may be used to eliminate the occurrence of power jackknife. For this reason it was desirable to re-examine devices about the fifth wheel to determine if a device which meets the performance criteria could be developed.

As shown on Figure 14, devices about the fifth wheel can be categorized as locking, damping or a combination of the two.

LOCKING

Locking encompasses all devices which physically restrain the articulation angle from changing when the onset of jackknife or trailer swing is sensed. As well, it includes devices which limit the articulation angle to a specific range during highway travel.

The advantages of locking the articulation at the very onset of jackknife or trailer swing are:

- the tractor trailer relationship is maintained, therefore the vehicle should continue on its intended path
- the vehicle may remain under control

The disadvantages are:

- the device when active does not allow normal steering
- the AJS resistive torques required are large
- the unlocked tires may not laterally support the vehicle (loss of vehicle control/stability)

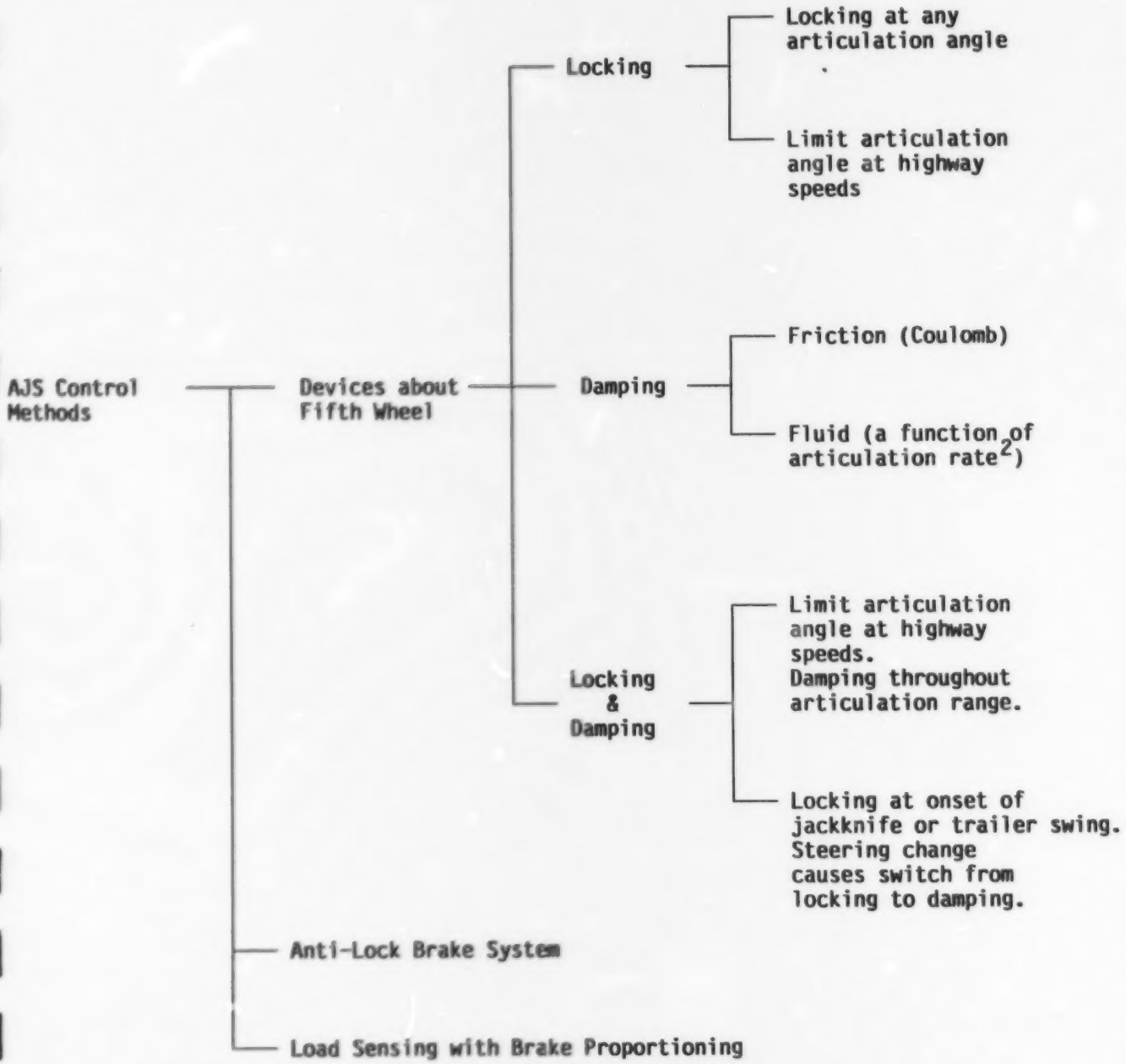

AJS Control Methods
Devices about Fifth Wheel
Locking
Locking at any articulation angle
Limit articulation angle at highway speeds
Damping
Friction (Coulomb)
Fluid (a function of articulation rate^2)
Locking & Damping
Limit articulation angle at highway speeds. Damping throughout articulation range.
Locking at onset of jackknife or trailer swing. Steering change causes switch from locking to damping.
Anti-Lock Brake System
Load Sensing with Brake Proportioning

Figure 14 Classification of AJS Control Methods

The advantages of limiting articulation to a specific range are:

- the tractor semitrailer relationship is maintained (tractor cannot jackknife into the semitrailer)
- the articulation limit (±16°) is possibly within a range that the driver could regain control of the vehicle
- the device does not require sensing

The disadvantages are:

- the jackknife or trailer swing will articulate to the limit every time, therefore potentially interfering with oncoming traffic (violates the fundamental performance criteria of an AJS)
- the lateral stability of vehicle may be reduced

DAMPING

Damping slows down the articulation rate during jackknife or trailer swing so that the driver has time to take corrective action and regain control of the vehicle. This method is primarily beneficial in tractor jackknifing, as it is characterized by very rapid articulation rates which exceed the best possible driver reaction time.

There are a number of methods to dampen the articulation rate, but they can generally be grouped as friction (coulomb) damping and fluid (viscous and turbulent flow) damping.

Friction damping is achieved through the use of a braking mechanism (i.e. calliper and disk, etc.) and can be either a constant resistive torque or a varying torque.

The advantages, in addition to providing the driver with more reaction time, are as follows:

- the device is simple in design, maintenance and use
- the device uses readily available components

The disadvantages are as follows:

- the device slows rather than stops the jackknife or trailer swing, thus failure of driver to react in the available time could result in a serious accident
- the device is costly to achieve even the minimum level of torque required

Fluid damping is achieved by restricting or controlling the flow of fluid. It can be classified as viscous or turbulent flow damping. Fluid damping can be termed a "passive" type device as it is always present and does not rapidly induce restraining torque only when jackknife has occurred.

Viscous damping is achieved under laminar flow conditions and the retarding force generated is proportional to the velocity of the device being slowed down (in this case, the articulation rate of the tractor semitrailer).

Turbulent flow damping generates a retarding force proportional to the velocity squared of the device being slowed down (in this case, articulation rate squared). Only turbulent flow (articulation rate squared) damping was examined in detail, because it provides considerably more resistance than viscous damping at high articulation rates.

The advantages of fluid damping are:

- the device creates little resistance at low articulation rates characteristic of steering changes, but it creates a large resistance at high articulation rates characteristic of jackknifing
- the device requires no sensing of jackknife/trailer swing onset if a constant damping coefficient is used

The disadvantages are:

- the damping coefficient is a compromise between the need for sufficient resistance at high articulation rates and the need not to impede steerability under normal operation.
- the device has little resistance to trailer swing as articulation rates are low - similar to those attained in steering manoeuvres.

LOCKING AND DAMPING

As locking may be preferable in some instances and damping in others, the combination of the two may result in a device with a better performance rating.

6.4.2 Investigation of the Design Feasibility of the Control Methods

Before evaluating these methods with the computer simulation, it was necessary to determine if such systems could actually be built and if they would be economically viable. That is, considering the space limitations around the fifth wheel, the capabilities of commercially available components and the cost, could an acceptable device be designed and built.

Articulation Locking

Investigations determined that there are three methods of obtaining or approaching a locking of the articulation during jackknife or trailer swing. These methods, all centred about the fifth wheel, are:

1) brake callipers and a disk

2) a toothed ram against a gear linked to the articulation

3) locking hydraulic cylinders linked to the articulation.

The brake calliper option needed a large calliper area (multiple callipers) to attain locking. As was discussed in Section 4.2, Brake Calliper and Disk, four of the largest callipers of current design would be required to lock the

articulation for an empty trailer jackknife. Due to the high cost associated with the callipers, this option was considered unattractive.

The toothed ram and gear (See Section 4.2 Locking of Articulation) has a large gear fixed to the fifth wheel and a toothed ram fixed to the trailer. At the onset of jackknife or trailer swing the ram was forced against the gear, locking the trailer to the tractor. This method appeared to have merit as it was fairly simple and the maximum resistive torque was solely a function of the strength of the components. However, there was concern that the ram might not engage properly with the gear if there was any movement between them and that the cost of a large gear and toothed ram capable of sustaining the expected loads would be prohibitive. If steering changes are required simultaneous with the jackknife condition this type of device, which is either on or off, would not be desirable.

The hydraulic cylinder option achieves locking of the articulation by blocking the flow of fluid from hydraulic cylinders linked to the articulation.

Due to restricted space around the fifth wheel and current hydraulic cylinder design limits the maximum torque attainable would be about 64,400 N·m (based on using two 10 cm (4 inch) bore hydraulic cylinders linked to a disk, on opposite sides, at the 23 cm (9 inch) radius point). The disk would connect by an interface to the trailer. The cylinders would have a stroke of about 46 cm (18 inch) to allow for sharp articulation angles as might occur in parking lot manoeuvres. Greater torques could be attained by using a larger diameter disk or by using larger bore cylinders; however, this would create mounting difficulty and require higher strength cylinders than are typically available.

This torque limit of 64,400 N·m was only sufficient to prevent tractor jackknife for the light or empty load conditions of a 14.6 m (48 ft) trailer. For heavier payloads and for trailer swing it was unable to prevent articulation. However, this level of torque should provide enough damping to allow the driver to react and take corrective action.

Of the two acceptable systems, only the hydraulic cylinder method had the flexibility of switching to damping from locking when required. As mentioned earlier, locking was not desirable when steering changes were performed during the AJS activation period. This was complimented by Susemihl's statement that active control at the fifth wheel would require a very large moment and would reduce the controllability of the vehicle (6). As well, the analysis had indicated that locking may cause vehicle instability. For these reasons the ability to switch to damping was very necessary and resulted in the selection of the hydraulic option.

6.5 Evaluation of AJS Control Methods

The different control methods were evaluated by means of the computer simulation. That is, the most promising control methods were modelled, using the design limitations of the previous investigation, and their performance

in controlling jackknife and trailer swing evaluated against the criteria of an ideal AJS.

The following discussion outlines the methodology and further details on evaluating the performance of the devices.

6.5.1 Methodology

A bottom up, or progressive test method was used to evaluate the various AJS methods. That is, the anti-jackknifing system was split into its subsystems of sensing and articulation control. Each of these subsystems was evaluated independently on simple manoeuvres such as a constant radius turn. If their performance was judged acceptable, the subsystems would then be combined to evaluate the total system performance. An acceptable subsystem performance was never achieved, thus total system performance testing was never reached.

In order to better evaluate the performance of the AJS control methods a performance range was established. This was done by having the vehicle configurations initially complete the specified manoeuvres without inducing jackknife or trailer swing (the ideal case) and then repeating the manoeuvre with jackknife and trailer swing induced, but without any AJS (the worst case). The performance of any AJS would have to fall in this envelope to register any improvement over a vehicle with no AJS. The rating of a device improved as the performance approached the "ideal case".

6.5.2 Performance Results

An analysis in the form of performance charts compared the effectiveness of each device against the performance criteria and performance envelope.

Locked Articulation

The first method evaluated was locking of the articulation. As described in the Design of an AJS (Section 6.4.2) it was not possible to achieve locking. Rather, a resistive torque of 64,400 N•m was the upper realistic limit achievable. For computer simulation this system was modelled as a constant torque of 64,400 N•m applied shortly before the wheels locked.

Constant Torque - Tractor Jackknife

From the performance chart (Table 7) it is evident that a constant torque of this magnitude was quite effective in controlling jackknife. At the manoeuvre design speed of 50 kph and the heaviest trailer payload, this torque was able to keep the change in articulation angle to 1.6° from onset of jackknife in a time span of 1.5 seconds. In fact, 3 seconds are required before the 15° articulation angle limit was reached. Vehicle intrusion into the neighbouring lane was also well within the one metre maximum for the 1.5 second time period. In addition, vehicle lateral stability, as judged by tractor yaw angle and position, was maintained throughout the application of the torque. For lesser trailer

Table 7 AJS Performance Chart - Constant Torque With Tractor Jackknife

Device: Constant Torque = ± 64,000 N·m
Vehicle & Manoeuvre Data:
Tractor Wheelbase: 5.6 m Manoeuvre: 90 m Radius Turn
Trailer Wheelbase: 14.6 m
Speed: 60 kph (Design Speed = 50 kph)

PERFORMANCE RESULTS (JACKKNIFING)					
Trailer Payload (kg)	$\Gamma <$ ± 15°	Change in Γ	Within* 1 Metre	Maximum Intrusion	Comments
0	Yes	0.2°	Yes	-	Excellent performance, jackknife eliminated, vehicle stability and trajectory maintained.
4,500	Yes	0.4°	Yes	-	Very good performance, jackknife eliminated, vehicle stability and trajectory maintained.
9,100	Yes	4.2°	Yes	-	Good performance, vehicle still jackknives but heavily dampened.
13,600 @ 50 kph	Yes	1.6°	Yes	-	Good performance, vehicle still jackknives but heavily dampened.
REFERENCE DATA (Performance Range)					
13,600 No Braking	Yes	0°	Yes	0 m	**IDEAL CASE**
13,600 No AJS	No	57.5°	No	1.1 m	**WORST CASE**

*As vehicle path differed even before application of brakes, deviation could only be based or vehicle orientation relative to its own path. It was assumed that driver input would have kept the vehicle path identical for each trial - at least up until moment of brake application.

payloads, where the likelihood of jackknifing was greater, the results were even better.

Simulation runs were also done at the threshold speed (60 kph) for the manoeuvre. At this speed only the heaviest trailer payload caused vehicle response to exceed the allowable levels outlined in the performance criteria.

Constant Torque - Trailer Swing

Examining the performance chart (Table 8) it was observed that the application of a constant torque of 64,400 N•m, as well as levels of 60% or 80% of this value, resulted in undesirable vehicle response. The tractor tires had insufficient lateral force capability to support the entire vehicle and as a result the entire vehicle spun out of control during trailer swing.

Constant Torque - Conclusions

It was evident from the results that the application of large resistive torques at the onset of tractor jackknife was a very effective means of jackknife control. However, the application of large resistive torques during trailer swing resulted in a situation worse than having no AJS at all. This mean that an AJS utilizing large resistive torques must be able to differentiate between trailer swing and tractor jackknife to ensure that large torques are only applied during tractor jackknife.

Another problem noted with the use of large resistive torques was that some manoeuvres may require steering changes simultaneous to being in a jackknife condition. If the articulation was locked or movement severely restricted, steering changes would be very difficult - if not dangerous. This meant that if a steering change was sensed simultaneous to a jackknife condition the AJS would have to select a lesser torque or articulation resistance. The results have shown that the lower the torque the less effective the AJS became.

It may be countered that severe restriction of the articulation was acceptable up until the driver was able to react. When he reacted it would usually be in a manner which removed the cause of the jackknife (i.e. he would release the brakes or switch to pumping action of the brake for the brake induced jackknife). This meant that the jackknife condition would probably have been removed and a decrease in torque or articulation resistance to allow steering would have no detrimental consequences on vehicle control.

It was concluded that an AJS, utilizing large resistive torques - approaching locking of the articulation, would be fairly complex in order to accommodate those situations in which large resistive torques are undesirable. In order to determine if a simpler but still effective system existed, other forms of AJS were investigated.

Table 8 AJS Performance Chart - Constant Torque With Trailer Swing

Device: Constant Torque = See Chart
Vehicle & Manoeuvre Data:
Tractor Wheelbase: 5.6 m
Trailer Wheelbase: 14.6 m
Speed: 60 kph (Design Speed = 50 kph)
Manoeuvre: 90 m Radius Turn

PERFORMANCE RESULTS (TRAILER SWING)					
Trailer Payload (kg) & Torque Applied	$\Gamma < \pm 15°$	Change in Γ	Within* 1 Metre	Maximum Intrusion	Comments
0 64,400 N·m	Not Relevant	0.2°	No	1.0 m	Tractor rotates with trailer. Worse than no AJS.
0 51,520 N·m	Not Relevant	0.3°	No	0.9 m	Tractor rotates with trailer. Worse than no AJS.
0 38,640 N·m	Not Relevant	1.1°	No	0.9 m	Tractor rotates with trailer. Worse than no AJS.
13,600 308,000 N·m	Not Relevant	0.0°	No	0.5 m	Tractor rotates with trailer. Worse than no AJS.
REFERENCE DATA (Performance Range)					
0 No Braking	Not Relevant	0	Yes	0 m	**IDEAL CASE**
13,600 No AJS	Not Relevant	15.3°	No	1.8 m	**WORST CASE**

*As vehicle path differed even before application of brakes, deviation could only be based on vehicle orientation relative to its own path. It was assumed that driver input would have kept the vehicle path identical for each trial - at least up until moment of brake application.

Breeze Device - Tractor Jackknife

The Breeze device, described in Section 4.2, had a resistive torque which increased as the articulation angle increased. This device was evaluated by the Ministry of Transportation of Ontario (MTO) in 1979/80 (5) and was given the best rating of the non ABS devices. As it was capable of fairly large torques, it was desirable to compare it with the constant torque method.

Examining the performance chart (Table 9) it was noted that the device was not able to limit the articulation angle to the 15° maximum in the 1.5 second time span for the heaviest load condition at the design speed. In fact, over 30° of articulation resulted. The vehicle deviation from the intended path was just over the allowable limit. If no corrective action was taken by the driver the vehicle reached roll-over tendencies at about 2.5 seconds from the onset of jackknife. At lesser payloads, where jackknifing was more likely to occur the device may perform acceptably.

Breeze Device - Trailer Swing

Examining the performance chart for the Breeze device with trailer swing (Table 10), showed a slight improvement in decreasing the articulation rate and the total articulation angle of trailer swing. Tractor yaw rate was increased somewhat but the unit seemed to retain some lateral stability. Vehicle deviation from the intended path was beyond the allowable limit.

Breeze Device - Conclusions

The Breeze device was not as effective as the constant torque device for jackknifing but it did provide the operator with a possible chance to recover from the jackknifing situation at lesser payloads (i.e. almost empty or empty). However, there was a greater hazard to oncoming traffic because of the greater out of lane deviation. For the trailer swing situation it provided some damping action, but whether this action was of any value was debateable. The MTO evaluation which tested the device for lane change on a real vehicle made no mention of any impedance to steerability as a result of this device.

Articulation Rate Squared Damping - Tractor Jackknife

Subsequent to evaluating the Breeze device, a form of fluid damping in which the resistive torque was proportional to the square of the articulation rate (Torque = Damping Coefficient • Articulation Rate Squared) was studied. This had the advantage of providing large resistive torques at high articulation rates characteristic of jackknifing and of providing little resistance at low articulation rates characteristic of steering. The result was a passive system which required no sensing of jackknife or trailer swing, unlike the Constant Torque and Breeze devices which were active systems. A device based on this principle has already been developed (see section 4.2 Fluid Damping - Mather Device) and was evaluated in the MTO computer simulation (5). The 5600 N•m maximum torque capability did not provide adequate torque and thus this device did not

Table 9 AJS Performance Chart - Breeze and Articulation Rate Squared Damping with Tractor Jackknife

Device: See Chart
Vehicle & Manoeuvre Data:
Tractor Wheelbase: 5.6 m
Trailer Wheelbase: 14.6 m
Speed: 50 kph (Threshold Speed)
Manoeuvre: 90 m Radius Turn

PERFORMANCE RESULTS (JACKKNIFING)					
Trailer Payload (kg)	$\Gamma < \pm 15°$	Change in Γ	Within* 1 Metre	Maximum Intrusion	Comments
13,600 BREEZE AJS	No	23.7°	Yes	-	Unacceptable performance, jackknife not prevented just dampened. Performance may be acceptable with lesser payloads.
0 $\Gamma^{**}2$ Damping AJS	No	19.2°	Yes	-	Unacceptable performance, jackknife not prevented just dampened.
0 Modified $\Gamma^{**}2$ Damping AJS	No	10.4°	Yes	-	May be acceptable.
REFERENCE DATA (Performance Range)					
13,600 No Brake Application	Yes	0°	Yes	0 m	**IDEAL CASE**
13,600 Jackknife Induced No AJS	No	57.5°	No	1.1 m	**WORST CASE**

*As vehicle path differed even before application of brakes, deviation could only be based on vehicle orientation relative to its own path. It was assumed that driver input would have kept the vehicle path identical for each trial - at least up until moment of brake application.

Table 10 AJS Performance Chart - Breeze and Articulation Rate Squared Damping with Trailer Swing

Device: See Chart
Vehicle & Manoeuvre Data:
Tractor Wheelbase: 5.6 m
Trailer Wheelbase: 14.6 m
Speed: 50 kph (Threshold Speed)

Manoeuvre: 90 m Radius Turn

PERFORMANCE RESULTS (TRAILER SWING)					
Trailer Payload (kg)	$\Gamma < \pm 15°$	Change in Γ	Within* 1 Metre	Maximum Intrusion	Comments
13,600 BREEZE AJS	Not Relevant	11.3°	No	0.8 m	Slight reduction in articulation rate. Small increase in tractor yaw rate.
0 Modified $\Gamma^{**}2$ Damping AJS	Not Relevant	8.9°	No	0.4 m	Some reduction in articulation rate. Slight increase in tractor yaw rate.
13,600 Modified $\Gamma^{**}2$ Damping AJS	Not Relevant	9.6°	No	0.9 m	Similar performance.
REFERENCE DATA (Performance Range)					
0 No Brake Application	Not Relevant	0°	Yes	0 m	**IDEAL CASE**
13,600 Trailer Swing Induced No AJS	Not Relevant	11.4°	No	1.6 m	**WORST CASE**

*As vehicle path differed even before application of brakes, deviation could only be based on vehicle orientation relative to its own path. It was assumed that driver input would have kept the vehicle path identical for each trial - at least up until moment of brake application.

receive a favourable performance rating. In this study the concept was given a much higher maximum torque capacity (65,000 N•m) and the damping coefficient was doubled.

Examining the performance results in Table 9 indicated this device was not capable of meeting all the performance criteria, for the vehicle configuration and manoeuvre simulated. The articulation angle had reached 27° in the 1.5 second time span from onset of jackknife, thereby exceeding the 15° limit. Although the vehicle had not deviated beyond the allowable path limit, it was very close to it. A further examination of the simulation data reveals that unlike a vehicle with no AJS the articulation rate increased to a certain level and then stayed fairly constant. If the damping coefficient were larger this level would be lower and it might be possible to meet the performance criteria.

The vehicle steerability was impeded to a greater and greater extent with increasing damping coefficient. Thus, a passive system, where the device is always on, would be undesirable if the damping coefficient was set too high. This investigation indicated that the damping coefficient used for this trial may very well be an upper limit for a passive system. The use of an active device, whereby this large resistance to articulation was only applied during jackknife or swing is one solution to the problem.

Modified Articulation Rate Squared Damping - Tractor Jackknife

As seen by the results of the constant torque method, large resistance during trailer swing caused vehicle lateral instability, yet, large resistances were required to effectively impede jackknifing. Two solutions to this problem were tested.

The first was to tailor the damping coefficient so that the AJS provided the maximum resistance possible to trailer swing without causing the tractor to lose its lateral stability. As the articulation rate during tractor jackknife was much higher than the rate during trailer swing, sufficient resistance may be created with this type of damping that the device will be effective for jackknifing as well. There were two ways to implement this solution. One way was to have a constant damping coefficient. However, it was doubtful that the coefficient could be set high enough to provide effective performance for trailer swing. It was also possible to have the damping coefficient increase with articulation angle. The results using the constant torque device and the Breeze Device showed that the resistive torque can increase with articulation angle without causing vehicle lateral instability. In fact, to obtain any kind of effective performance against trailer swing this increase was essential.

The second solution, which was more complex, required that the AJS distinguish between jackknife and trailer swing. It could then apply the appropriate level of resistance to each.

The first solution was studied utilizing a damping coefficient which increased with articulation angle [Torque = $(K + K_1 \Gamma) \cdot \dot{\Gamma}^2$] where $K + K_1$

are constant. This variance in effective damping coefficient ($K + K_1\Gamma$) lead to the expression 'variable damping device'.

Examining the performance chart for jackknifing (Table 9) it was seen that the device had allowed the articulation to exceed the allowable limit of 15° by 3.25° in the 1.5 second time span for the zero payload condition. Further increases in articulation angle occurred very slowly and it was quite reasonable to assume that the driver would still be able to take corrective action and regain control of the vehicle. Examining vehicle deviation from the intended path it was seen that the vehicle did not exceed the limit but came quite close to it.

Modified Articulation Rate Squared Damping - Trailer Swing

The performance of this device with trailer swing (Table 10) was very similar to that of the Breeze device. That is, the articulation rate was slightly reduced (10 to 20% less than no device at all) and the maximum articulation angle was somewhat less (36° vs 41° for no device).

Modified Articulation Rate Squared Damping - Conclusions

It was clear that increasing the damping coefficient in proportion to the articulation angle improved the performance of a device using this concept. If the device was able to distinguish between jackknife and trailer swing the performance would be even better. However, the requirement of sensing the onset of jackknife or trailer swing removed the major attraction of this concept: simplicity due to the passive operation.

Articulation Limiting (Safe-T-King) - Tractor Jackknife

Examining the performance chart (Table 11) it was noted that for the long trailer and long tractor combination (Combination 1) the performance was acceptable. The jackknife angle did not exceed 15° and the vehicle seemed to remain on the road. Vehicle deviation from the intended path was within the allowable range of 1 m. The combination of short, empty trailer with the long tractor (Combination 2) caused the entire vehicle to rotate.

Articulation Limiting (Safe-T-King) - Trailer Swing

The performance chart (Table 12) indicated that, for both the empty and highest payload conditions, the tractor rotated with the trailer once the articulation limit was reached. This was not acceptable.

COMPARISON AND SUMMARY OF THE DEVICES

In order to better illustrate and compare the effectiveness of the various control methods, the vehicle responses for all devices were combined to form a series of plots (Figures 15 and 17). These plots showed articulation angle, articulation rate, tractor yaw angle and tractor yaw rate versus time. The time span on the plots showed vehicle response for about 2 seconds after initiation of jackknife or trailer swing. Any values beyond this time would be affected by driver response which is beyond the scope of

Table 11 AJS Performance Chart - Articulation Limiting with Tractor Jackknife

Device: Articulation Limiting (Safe-T-King)
Vehicle & Manoeuvre Data:
Tractor Wheelbase: 5.6 m
Trailer Wheelbase: See Chart
Speed: 50 kph
Manoeuvre: 90 m Radius Turn

PERFORMANCE RESULTS (JACKKNIFING)					
Trailer Payload (kg)	$\Gamma < \pm 15°$	Change in Γ	Within* 1 Metre	Maximum Intrusion	Comments
Long Trailer 13,600 kg	Yes	7.3°	Yes	-	Fairly good performance.
Short Trailer 0 kg	Yes	10.7°	No	0.5	Trailer rotates with tractor.
REFERENCE DATA (Performance Range)					
13,600 No Brake Application	Yes	0°	Yes	0 m	**IDEAL CASE**
13,600 Jackknife Induced No AJS	No	57.5°	No	1.1 m	**WORST CASE**

Table 12 AJS Performance Chart - Articulation Limiting with Trailer Swing

Device: Articulation Limiting (Safe-T-King)
Vehicle & Manoeuvre Data:
Tractor Wheelbase: 5.6 m
Trailer Wheelbase: See Chart
Speed: 50 kph
Manoeuvre: 90 m Radius Turn

PERFORMANCE RESULTS (JACKKNIFING)					
Trailer Payload (kg)	$\Gamma < \pm 15°$	Change in Γ	Within* 1 Metre	Maximum Intrusion	Comments
0	Not Relevant	11.4°	No	0.5	Some rotation of tractor with trailer.
REFERENCE DATA (Performance Range)					
13,600 No Brake Application	Not Relevant	0°	Yes	0 m	**IDEAL CASE**
13,600 Trailer Swing Induced No AJS	Not Relevant	11.4°	No	1.6 m	**WORST CASE**

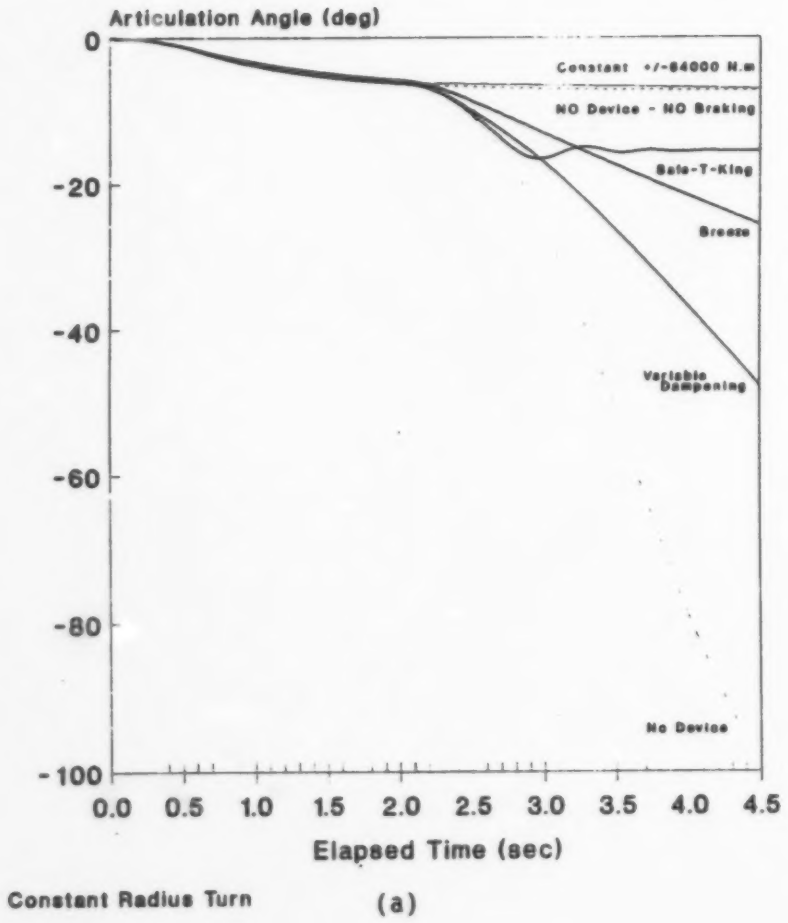
computer program's capabilities. To supplement these plots, graphical representations of a tractor semitrailer outfitted with the devices were constructed (Figures 16 and 18). The heaviest payload case presented the most demanding scenario for an AJS. For this reason it was used to construct the diagrams.

The dotted lines in Figure 15 indicate the performance envelope into which the devices must fall for the tractor jackknife situation (ideal and worst case). The label 'worst case' is used to indicate the occurrence of jackknife without an AJS fitted to the vehicle, and conversely 'ideal case' indicates the absence of jackknifing. The closer the device brings the vehicle to its ideal characteristics (no jackknife), the better it was rated.

Examination of Figure 15 shows that the constant torque device (64,400 N•m) is the most effective control method as it quite closely corresponds to the ideal situation. The articulation characteristics are almost identical to the ideal case, while the tractor yaw values differ slightly. This tractor yaw variance can be explained by realizing that the constant torque is applied prior to rear tractor tire lock-up. Evident from the articulation angle and articulation rate plots, the articulation is effectively locked. The variance in tractor yaw must therefore be accredited to the locking of the rear tractor tires. Simulation results showed that the lateral forces on the tires remain saturated from the point of lock-up (reduced from the ideal case of ≈ 3,000 N to 100 - 200 N) for the duration of the run. The front tires, 1 and 2, are thus forced to compensate and the lateral forces increase. This increase in lateral force increases the slip angle of the tires which results in the change in vehicle heading. This explains the tractor yaw variance and the approximate 1 metre deviation in the tractors path shown in Figure 16(c).

The Safe-T-King (articulation limiting device) displays adequate tractor jackknife control. The articulation limit of 15° is reached as quickly as the no device case, but is never exceeded. Examination of all four plots in Figure 15 shows that with the Safe-T-King a damped "bouncing" type of response results when the articulation limit is reached. This is due to impact of the secondary king pin with the urethane plug at the limit of the grooved slot. This bouncing is damped due to energy absorption of the urethane plug and the tires. Tests conducted by MTC in 1979/80 (5) also showed this occurrence and as a result rated the device as fair for maintaining driver control. The orientation and position of a tractor semitrailer, fitted with the Safe-T-King, involved in a jackknife on a roadway is shown in Figure 16(f).

Variable damping also displays adequate tractor jackknife control. The rate of articulation divergence is initially less than the Safe-T-King and remains approximately constant. The difference in articulation angle between the Safe-T-King and variable damping device is only 2 to 3 degrees at the 1.5 second mark from the point of tire lock-up. However, the articulation angle continues to increase, at a constant rate, to reach a maximum of 25° in 2.3 seconds after lockup. The tractor yaw divergence



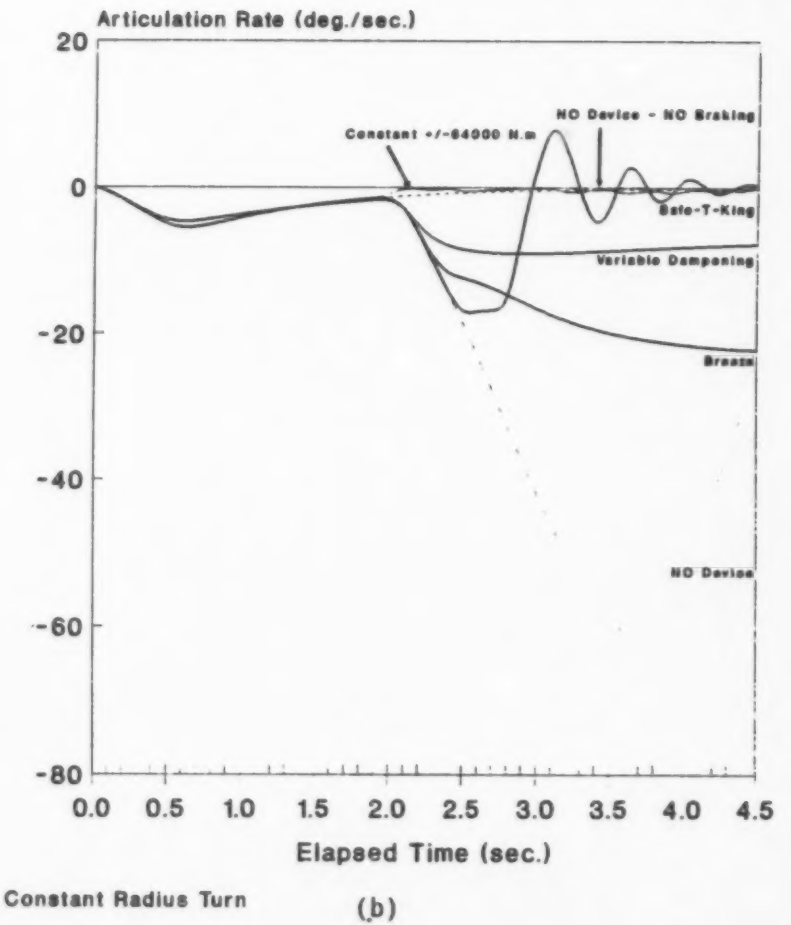
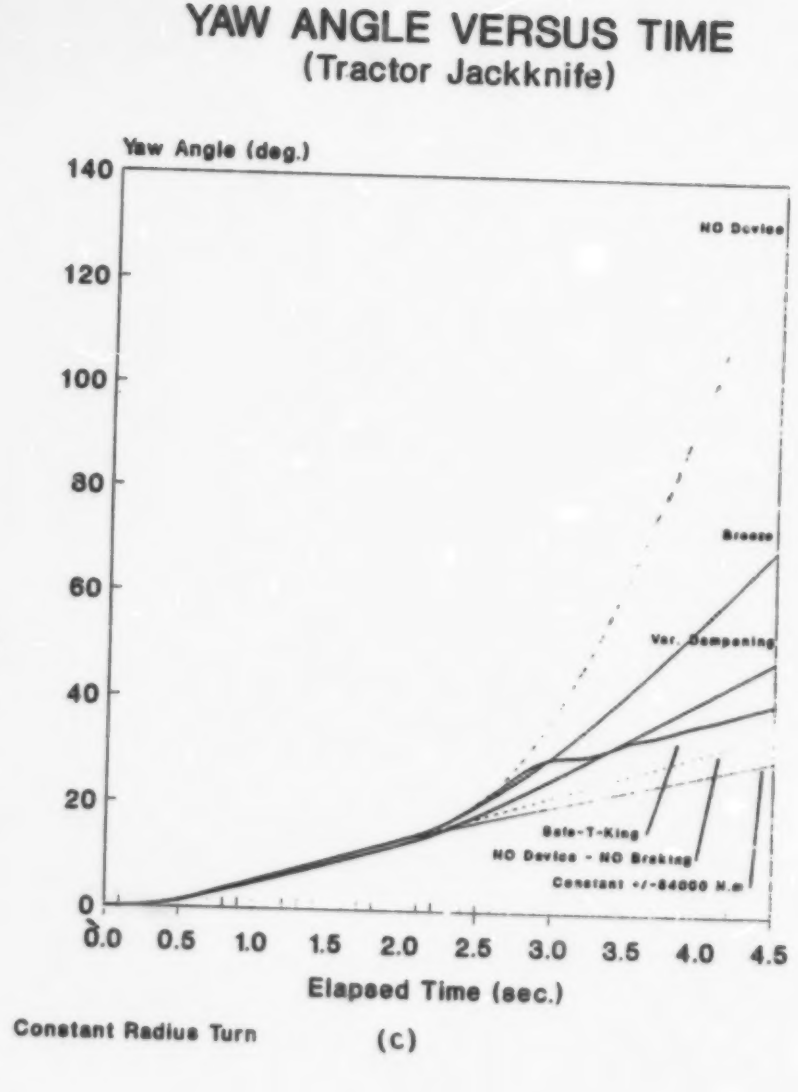


Figure 15 Relative Performance Plots of the Devices Used to Inhibit Jackknife



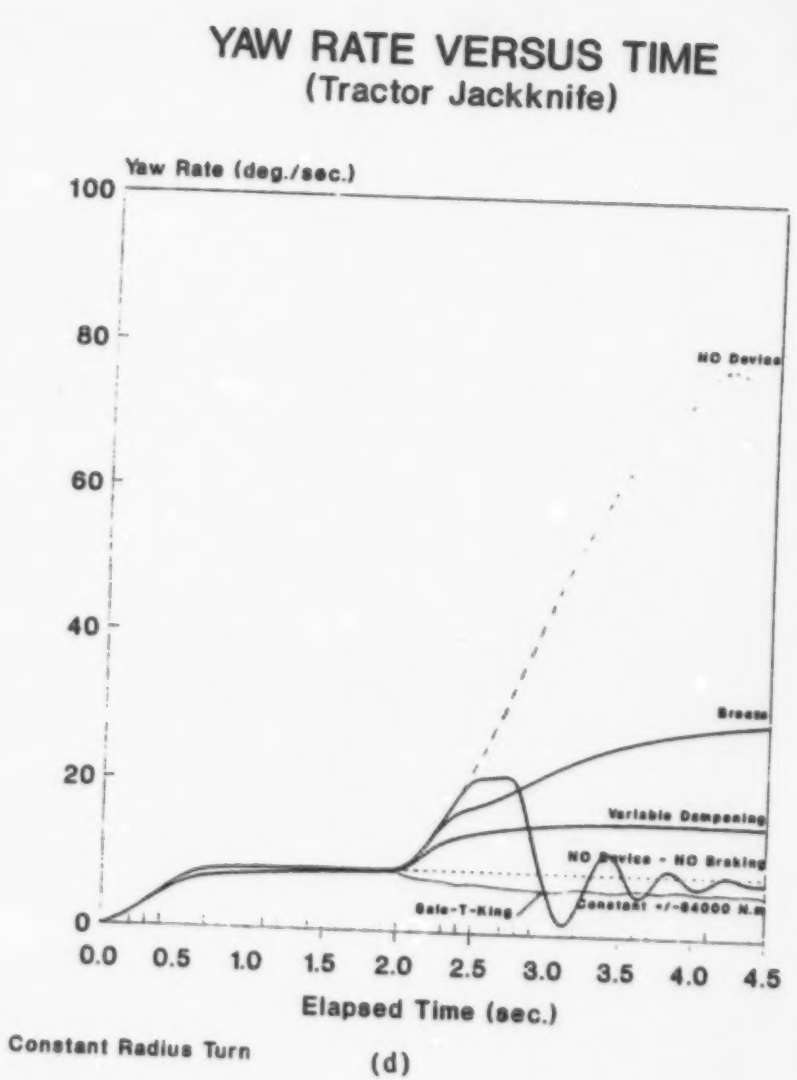
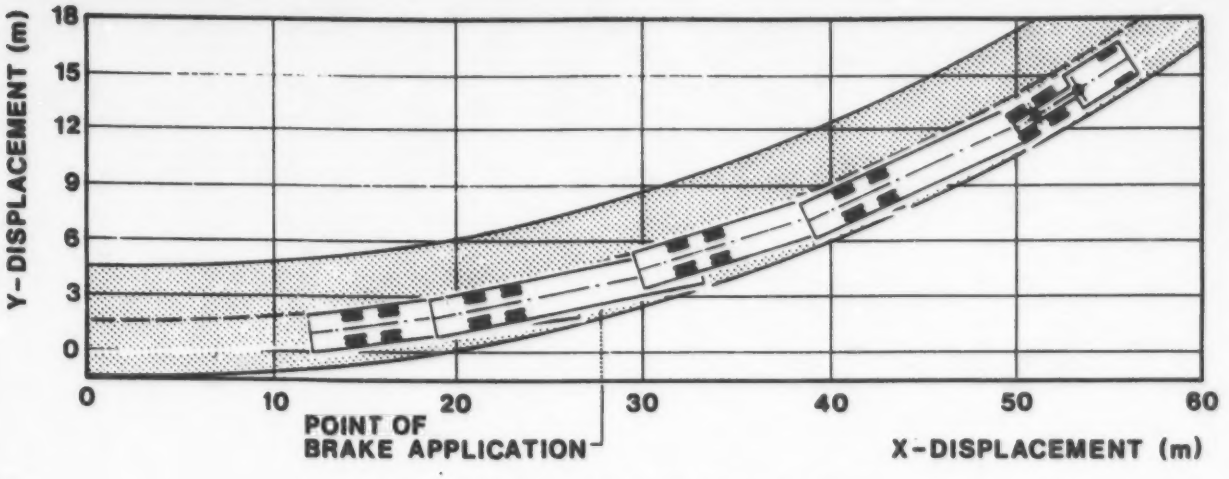
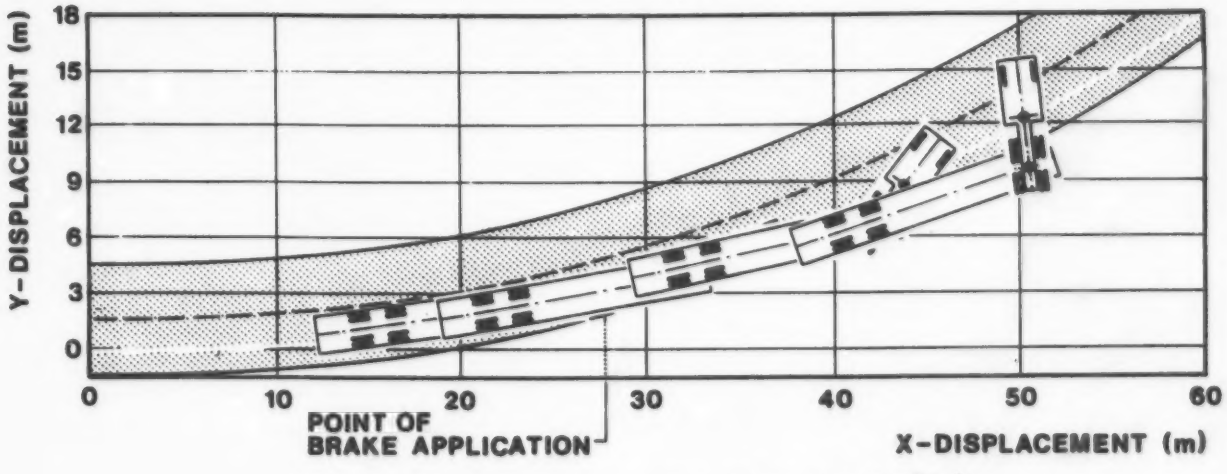


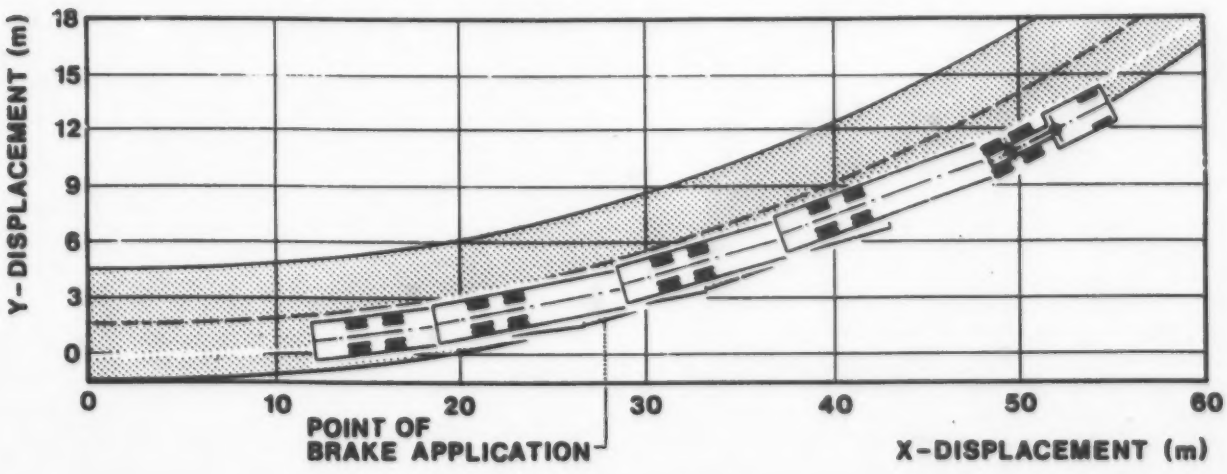
Figure 15 Relative Performance Plots of the Devices Used to Inhibit Jackknife (continued)



(a) Ideal Situation (no braking)

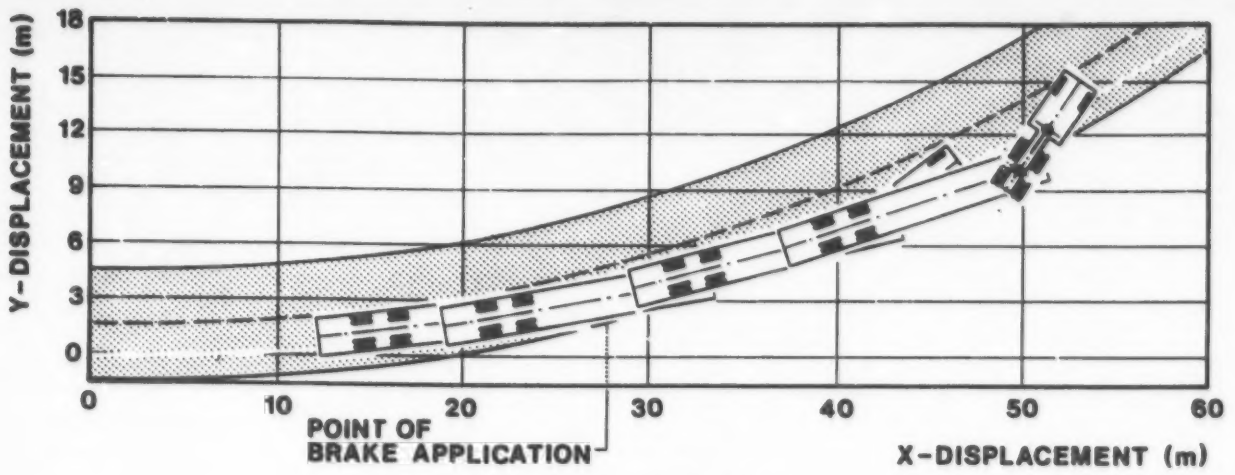


(b) Locking of Rear Tractor Tires (no device)

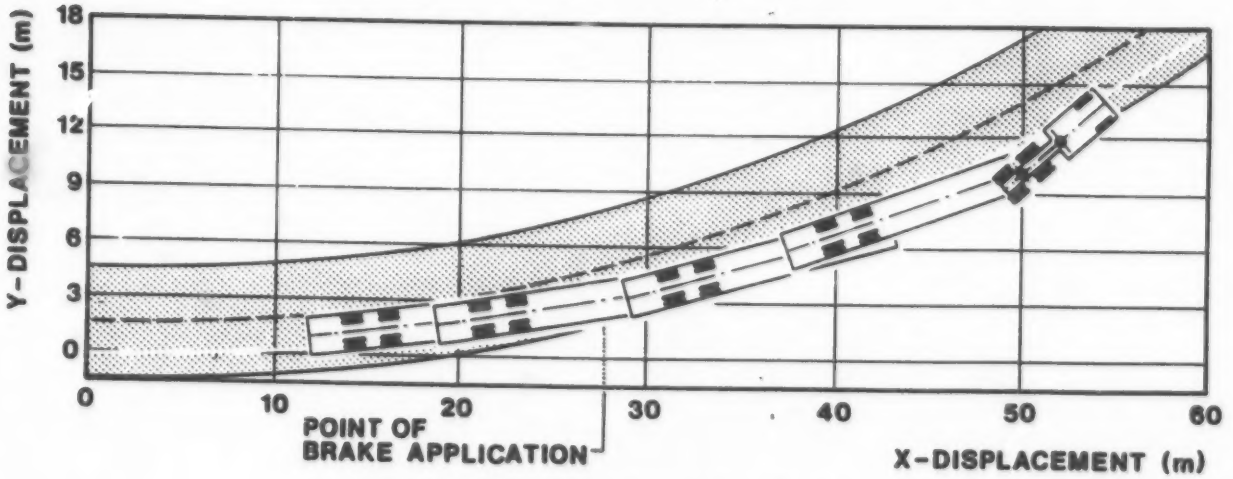


(c) Constant Torque Device

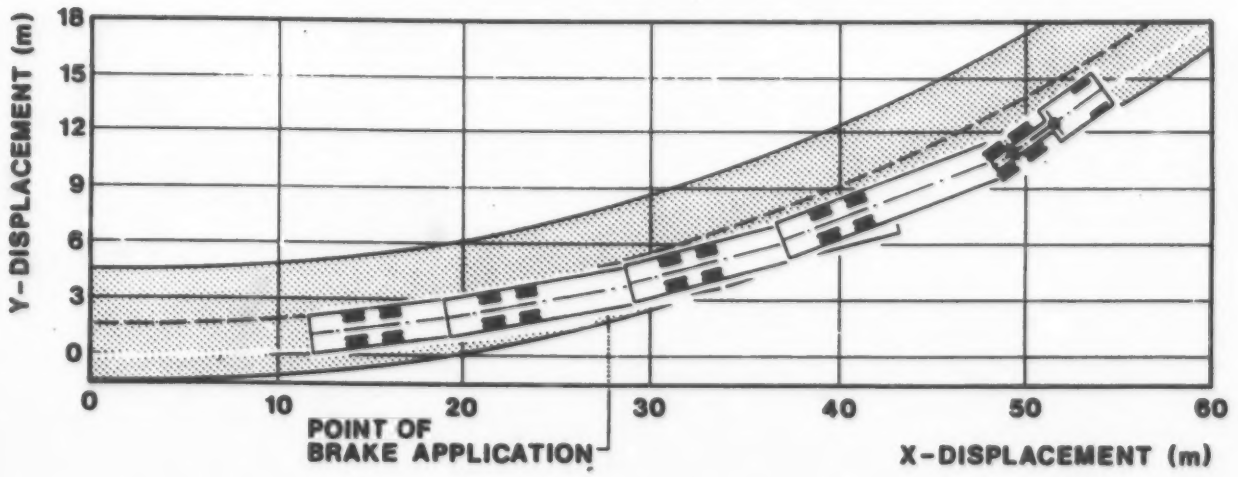
Figure 16 Path of a Tractor Semitrailer in a Jackknife Situation Fitted With Various Anti-Jackknife Devices



(d) Breeze Device



(e) Variable Dampening Device



(f) Safe-T-King Device

Figure 16 Path of a Tracktor Semitrailer in a Jacknife Situation Fitted With Various Anti-Jackknife Devices (continued)

behaves in a similar manner and results in the vehicle deviating even more from its intended path as shown in Figure 16(e).The Breeze device, overall, did not perform as well as the other devices examined, but still showed considerable improvement over the 'worst case' condition. This is evident from all plots in Figure 15 and the graphical representation in Figure 16(d).

Figures 17 and 18 confirm the results already presented in the performance charts for trailer swing (Tables 8, 10 and 12). All devices have either a minimal or negative impact on vehicle response. For example, Figures 17(a) and 17(b) shows that the constant torque application improves the articulation characteristics of the vehicle. In contrast, Figures 17(c) and 17(d) show that the constant torque application has a very negative impact on the tractor yaw characteristics. This response is undesirable as the tractor is forced to rotate out of control with the trailer. Figure 18(e) shows how this uncontrollable action directs the tractor into the neighbouring lane of oncoming traffic (approximately 1 m at the point indicated) and the trailer swings off the roadway by approximately 2 m. The other devices behave in a similar undesirable manner. As was previously discussed, the 1.5 second reaction time of the driver was used as a time frame in which the devices were evaluated. Their relative performance in this time period has just been discussed, but performance outside this window remains to be presented. Appendix "G" contains plots similar to Figures 15 and 17 that show performance results, of the devices discussed, in a time span of 7.5 seconds from the point of tire lock-up.

6.6 Influence of Driver Response on AJS Results

As it was difficult to account for driver reactions to the jackknife or trailer swing with the computer simulation, it was important to analyze the impact that this reaction would have on the performance results presented.

There are essentially two driver responses. Firstly, he can, where he is the initiator of the jackknife or trailer swing, remove the cause (i.e. switch from constant brake application to a pumping action for the brake induced jackknife, depress the clutch for a power jackknife). Secondly, and probably subsequent to the first, he can take some corrective action to get his vehicle back into its intended path. This would probably mean some form of rapid steering reversals.

It was possible to approximate the first driver response in the simulation by completely removing the brake application at about 1.5 seconds after the initiation of the jackknife. The first trials were done with an AJS in place (constant torque = ± 64,400 N·m) and the second without an AJS. All trials were performed on a constant radius turn.

With the AJS in place, releasing the brakes caused the jackknifing process to stop. With the constant torque of ± 64,400 N·m the change in articulation over the 1.5 second time span was minimal and the trial results when viewed against the performance criteria were said to be excellent. In addition, the trial was conducted with the largest payload, which indicated that the results would be even better at lower payloads, where the probability of

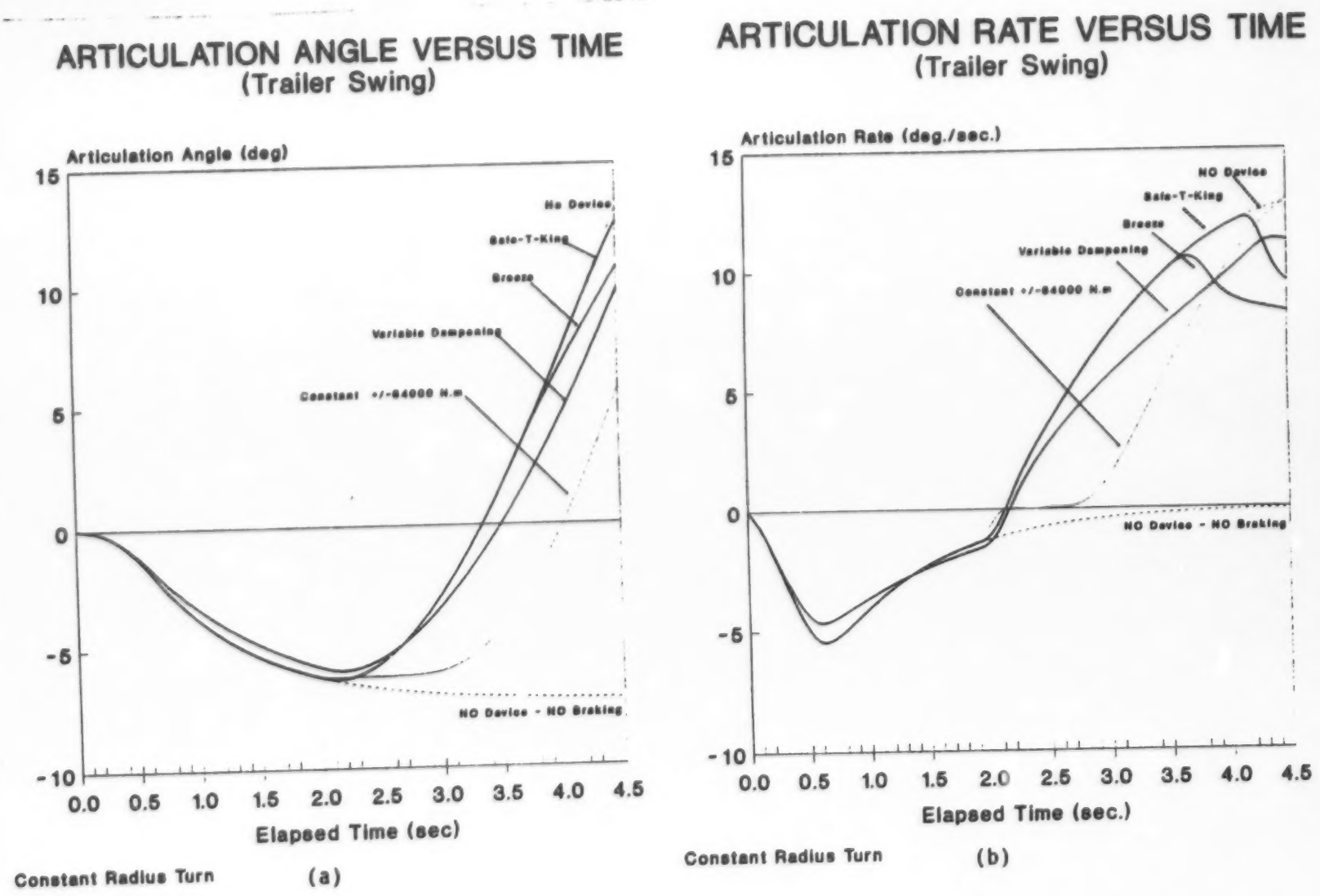
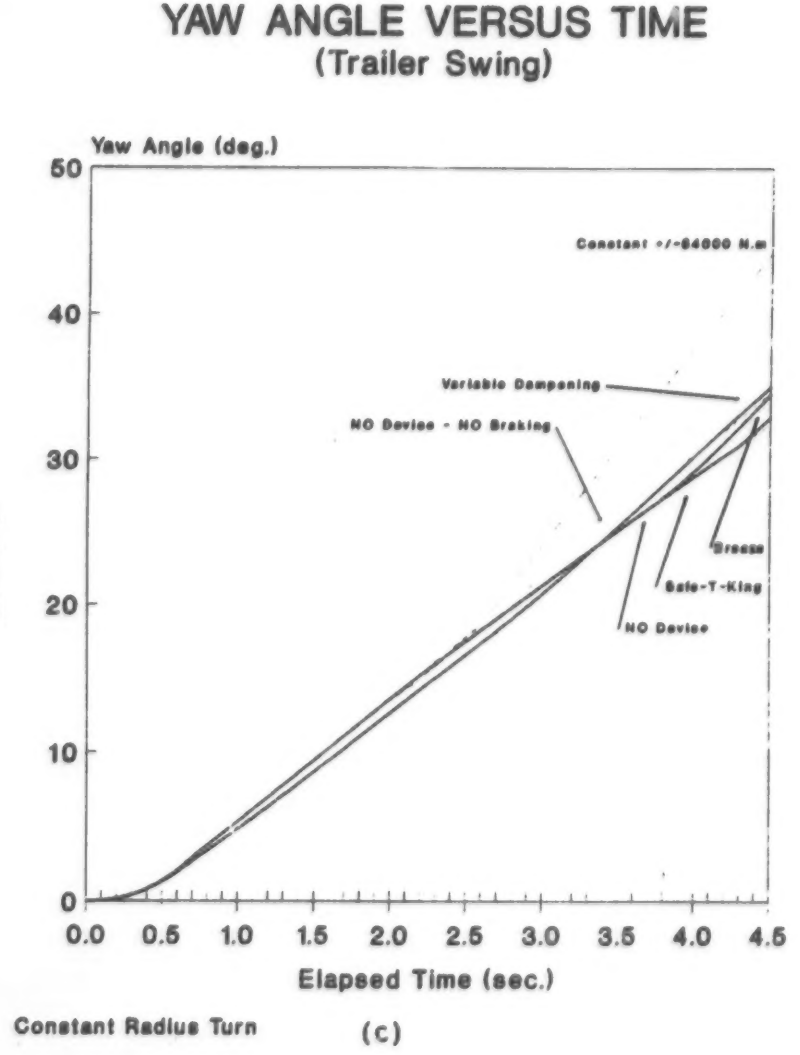


Figure 17 Relative Performance Plots of the Devices Used to Inhibit Trailer Swing



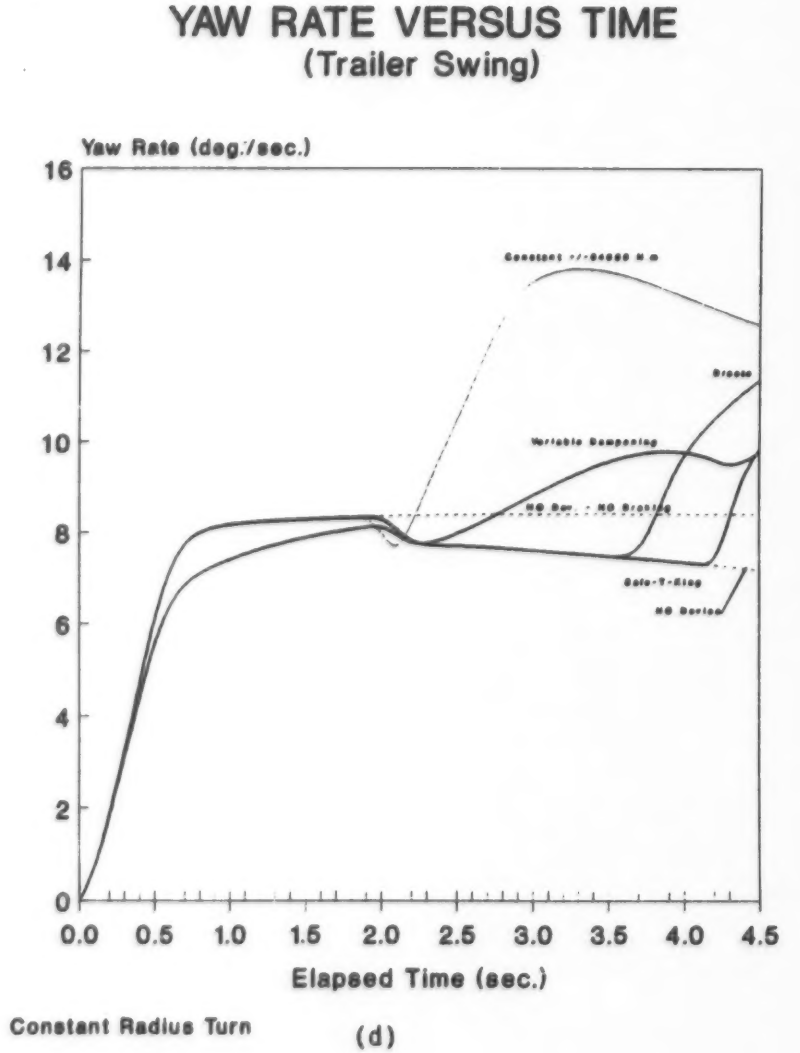
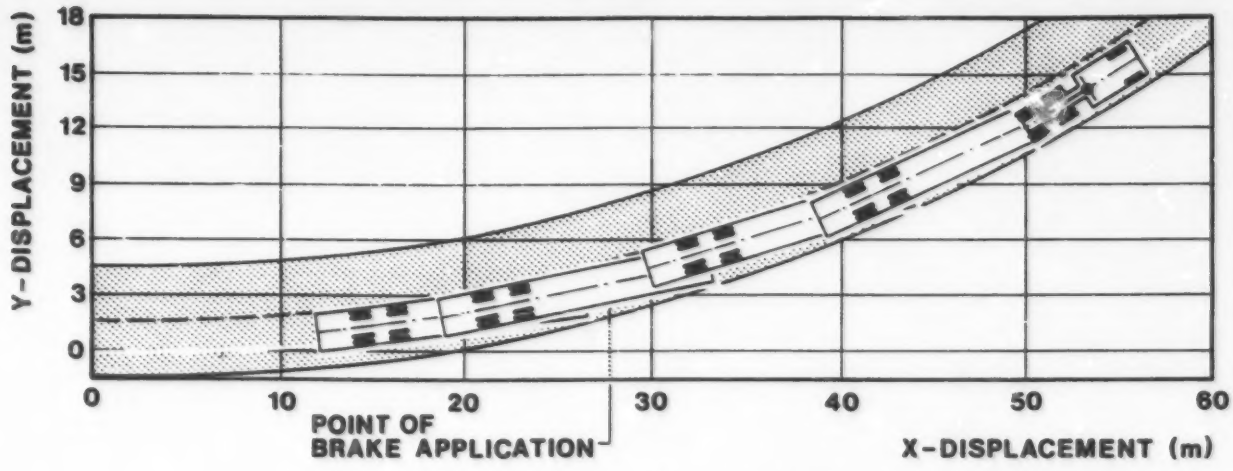
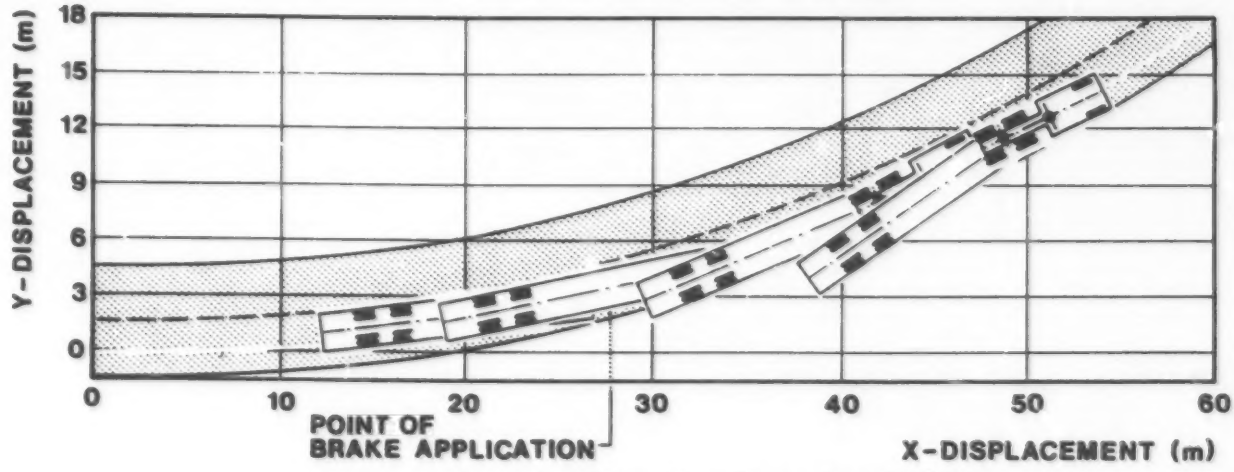


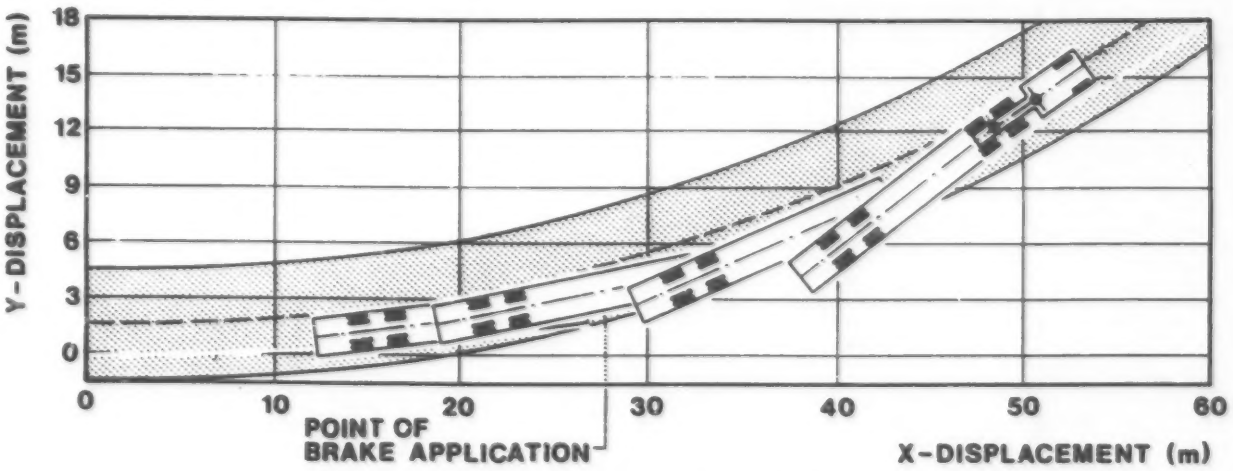
Figure 17 Relative Performance Plots of the Devices Used to Inhibit Trailer Swing (continued)



(a) Ideal Situation (no braking)

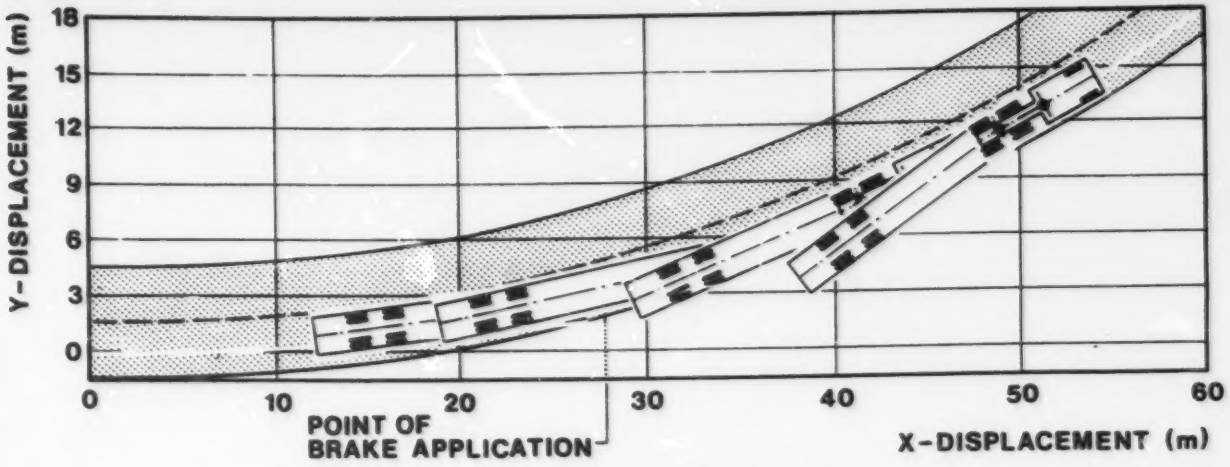


(b) Locking of Semitrailer Tires (no device)

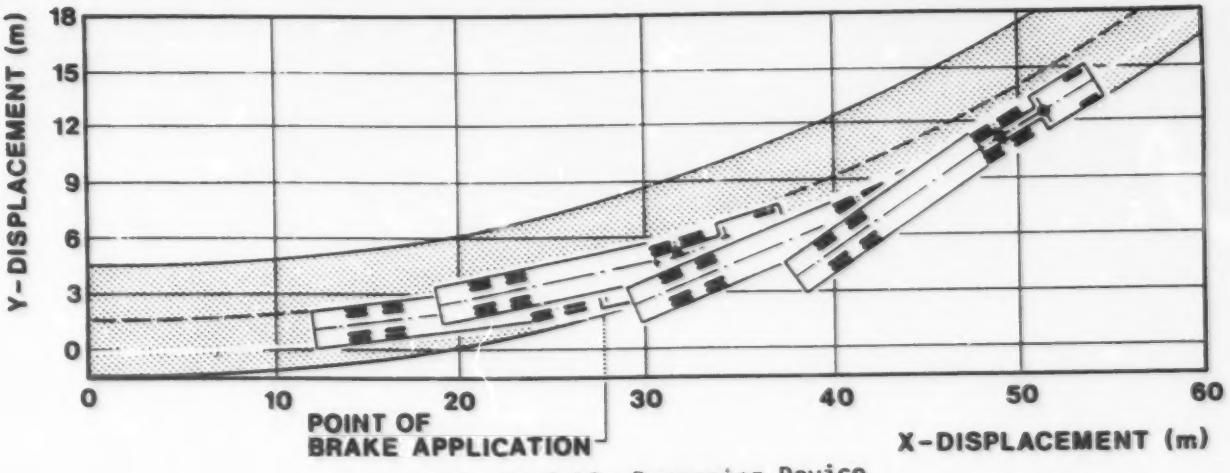


(c) Constant Torque Device

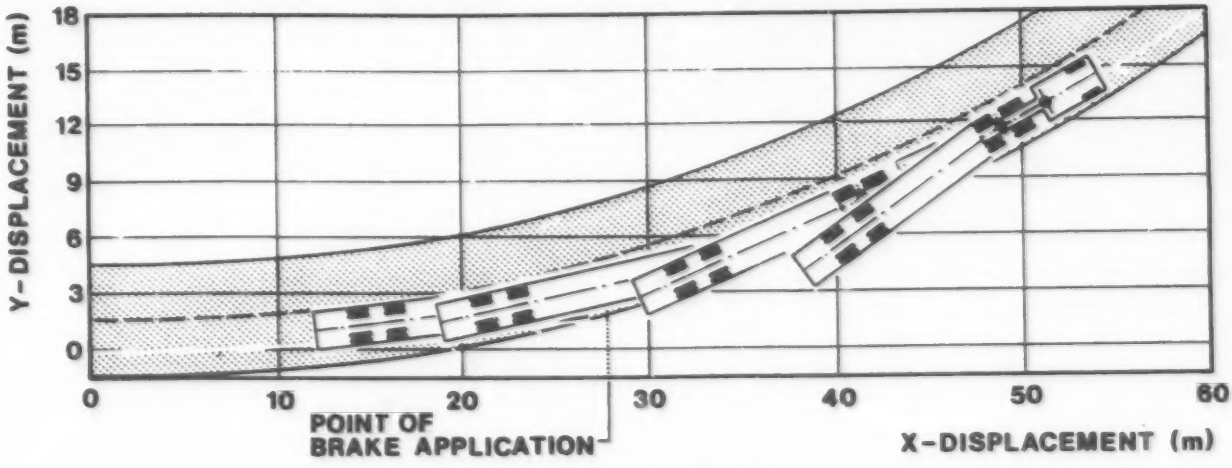
Figure 18 Path of a Tractor Semitrailer in a Trailer Swing Situation Fitted with Various Anti-Jackknife Devices



(d) Breeze Device



(e) Variable Dampening Device



(f) Safe-T-King Device

Figure 18 Path of a Tractor Semitrailer in a Trailer Swing Situation Fitted with Various Anti-Jackknife Devices (continued)

jackknife is greater. The fact that the driver would probably switch to a pumping action of the brakes rather than completely releasing them should make little difference to the results.

Where no AJS was in place, releasing the brakes did not cause the jackknifing action to stop; the rate was only slightly reduced.

The second driver response is the corrective action to get the vehicle back on its path. As demonstrated, if the AJS has sufficient resistive torque, the vehicle deviation from its intended path is minimal and corrective action may not be required. However, there are some vehicle manoeuvres in which the AJS can not apply large torques and as a result the jackknife will become more severe before it is apprehended. This means that the driver will be required to perform rapid steer reversals to get the vehicle back under control. This type of driver response is not desirable as it means that the vehicle may cross several lanes before the driver regains control or that the driver input may worsen an already detrimental situation.

Similar conclusions have been made by other authors. MTC in their 1979 study (5), performed tests which allowed the driver to try and avoid an impending jackknife by steering in a corrective manner. Results showed that an additional 5 or 6 lanes were required to regain control. Susemihl (6) assumed that corrective steering was not practical because of the resulting loss of driver steering control.

For trailer swing, the only correct driver response is to release the brakes or switch to a pumping action of the brakes. Steering changes have a minimal effect on the swinging action.

6.7 Influence of ABS and Front Axle Brakes on AJS Results

Legislation requiring front axle brakes on tractors in Canada is pending. As of the date of this report, it is anticipated that legislation will require brakes to be fitted on all axles such that the vehicle has an energy absorbtion capacity proportional to its weight. The vehicle will be required to meet specified deceleration rates and there will be no stipulation on the front to rear brake level proportioning. Because of this pending legislation it became necessary to evaluate the impact of front axle brakes on the performance of an AJS.

There is a trend in the trucking industry toward accepting ABS, with the realization that closer evaluation is still required. As described earlier, ABS in theory can improve the average traction output of a tire (both laterally and longitudinally) over that produced by a locked wheel. Recent advances in ABS technology, in the area of spin control, may also prove effective for prevention of most forms of power jackknife. Research into the state of the art technology of ABS was not within the scope of this project; ABS was only considered with respect to how it influenced the operation of anti-jackknife devices.

The antilock brake system modelled in this simulation program neglects the rotational dynamics of the wheels. Instead, it is based on slip which is calculated on a quasistatic basis (see Appendix "D"). This very much simplified model produces results that approach ideal and for this reason results should be interpreted with caution.

It can be surmised, since the two systems (ABS and AJS) operate completely independent of one another, that there should be no detrimental impact upon the results if both are operating simultaneously. In fact, the performance can only be improved as the ABS will lower the torque requirements of the AJS.

A complete investigation of the effect of front tractor brakes on the AJS performance results was not conducted. However, some assessment of the impact of front wheel brakes, based on the analysis, was made. Simulation runs in which jackknife and trailer swing were induced without an AJS but with front brakes were performed.

From the analysis of tractor jackknife, it was determined that when jackknifing occurs the tractor rotates about the front wheels. This is due to the greater lateral friction force on the front tires than at the rear tractor tandem wheels. If this friction force could be reduced, the jackknifing moment about the fifth wheel would be reduced. To verify this, simulation runs with light and heavy front brake applications were conducted. The simulations with a light level of front brake application showed a slight decrease in the jackknife rate. When the front wheels were heavily braked so as to lock the wheels the jackknife rate was significantly reduced (see Plots in Appendix "H"). This is as expected as locking reduces the lateral friction force at the front wheels and the entire vehicle continues on in the direction of travel.

The effects of ABS were also investigated. The plots in Appendix "H" show the results obtained. As was previously mentioned, the ABS model coded in the software is oversimplified and results should be regarded with caution. Discussions held with steering committee members concluded that a vehicle fitted with ABS under certain circumstances may still experience a brake induced jackknife.

7.0 CONCLUSIONS

The goal of this study was to determine the feasibility of an anti-jackknife system for tractor semitrailers used to restrain both jackknife and trailer swing.

This investigation resulted in the following conclusions:

1) Tractor jackknife and trailer swing are significant problems associated with tractor semitrailers. Six to eight percent of all combination unit accidents are the result of jackknife. Over 9% of these jackknife accidents result in fatalities and extensive property damage.

2) Enough concern has been generated by jackknife that hundreds of patents, designed to combat this hazard, have been registered. Only a few of these devices have been developed, and very few have gained any acceptance at all by the trucking industry due to industry doubts as to whether the devices function as intended.

3) Some of the devices that have been marketed to date are activated by brake pedal application and therefore are only effective with brake induced jackknife or trailer swing. As well, a number of these devices are only compatible with certain styles of trailers and require manual connect/disconnect at each trailer transfer and are thus easily neglected or forgotten.

4) The acceptance of an AJS by the trucking industry depends on two primary factors: 1) Cost and 2) Proven Performance. The proven performance requirement is a result of the flood of unproven and/or nonfunctional "add-on" safety devices that have entered the marketplace, creating scepticism in the industry.

5) Examination of the jackknife phenomenon indicated that tractor jackknife and trailer swing result from saturation or limitation of the lateral forces at the tire road interface. This saturation is usually caused by excessive brake application, usually to the point of wheel lock up. In the case of tractor jackknife, saturation of the lateral side forces at the tire/road interface may also be due to excessive throttle application, improper selection of a low gear on downhill descents, or an engine braking device.

6) Tractor jackknife and trailer swing behave in dissimilar manners on flat curves. The articulation angle in jackknife diverges more quickly and in an opposite direction than in trailer swing. Trailer swing proved to be a milder form of instability which tends to regain stability without any form of compensation.

7) Fifteen degrees of articulation may be used as an indicator for both jackknife and trailer swing. However, it does not indicate a "point-of-no-control" for trailer swing.

8) Sensing of the onset of jackknife or trailer swing (necessary for active devices which apply articulation resistance during jackknife or trailer swing only) is difficult because the characteristics of each are often duplicated in the transient responses of the vehicle during normal manoeuvring.

9) Sensing the onset of jackknife or trailer swing is possible by monitoring the relationship between certain characteristics or parameters and has been successfully implemented on articulated buses. Unfortunately, these characteristics are vehicle dependent and therefore do not readily lend themselves to tractor semitrailer applications where the configurations change on a regular basis.

10) There are a number of factors that will influence the severity of jackknife and trailer swing, as they are both configuration and payload dependent. An increase in inertia of either component, tractor or semitrailer, will slow the jackknife or trailer swing action because of inertial resistance. Increased inertia will also generate the requirement for a larger torque about the fifth wheel. Conversely, lower inertia causes a more rapid jackknife or trailer swing and generates lower torque values about the fifth wheel. Any increase in vehicle velocity will increase the severity of jackknife or trailer swing.

11) Large torques would be required to prevent further articulation of a tractor semitrailer during jackknife or trailer swing. These torques, for the vehicles studied, range from 52,000 N·m during tractor jackknife to 98,000 N·m during trailer swing for the empty configurations studied. The loaded configurations produce larger torque values that range from 93,000 N·m during tractor jackknife to 310,000 N·m during trailer swing.

12) Methods of attempting to control jackknife or trailer swing by controlling the articulation can be classified under the categories of locking, damping, limiting articulation or a combination thereof. Locking prevents any further articulation at the onset of jackknife or trailer swing. Damping, which can take many forms, consists of slowing down the articulation thereby providing the driver with more time to react. Limiting restricts the articulation range during highway travel.

13) All four devices modelled and presented in this report show merit for controlling tractor jackknife for the configurations studied. The relative performance of the devices in the 1.5 sec driver reaction time may be rated in the following order:
    1. constant torque device,
    2. Safe-T-King device (Articulation limiting),
    3. Variable damping device, and
    4. Breeze device (friction application).

Although the devices showed improvement in jackknife situations, all had problems associated with their use that require further development:

a) Constant Torque:
   - sensing is required.
   - effectiveness is reduced with increased payload.
   - steering is impeded when activated.

b) Safe-T-King:
   - limit is always reached.
   - control may still be lost in some situations.

c) Variable Damping:
   - damping coefficient required is large.
   - damping coefficient impedes steering if too large.

d) Breeze Device:
   - torques are generally not large enough for a significant effect.

14) The results of this study are consistent with the results of the principally experimental work conducted by the Ontario Ministry of Transportation in conjunction with TDC (5).

15) Anti-jackknife devices can generally be classified as "active" or "passive" devices. Active devices require sensing to indicate that jackknife or trailer swing has started and restraining torques may then be applied. Passive devices, such as the variable damping device in this study, are always applying resistive torque based on current conditions such as articulation rate, and thus tend to prevent the development of jackknife or trailer swing. Ideally, ABS and anti-slip control systems monitor and control wheel spin and may prevent the development of jackknife.

16) Except for ABS, the devices studied show little or no merit in controlling trailer swing. In some situations conditions are worse than if no device is applied. These adverse situations are due to the large torques produced during trailer swing. When trailer swing is inhibited, the rear tractor tires become saturated and the entire vehicle rotates out of control. This, combined with evidence that trailer swing is the cause of fewer reported accidents, indicates that fifth wheel devices used to control trailer swing may not warrant further study.

17) The key tool used to investigate the feasibility of an AJS about the fifth wheel was computer simulation. Many of the conclusions and findings are based on results obtained from the program used and are only as accurate as the model the program is based upon. For this reason, conclusions related to simulation were based on a relative comparison to avoid dependency on the absolute numbers generated.

8.0 RECOMMENDATIONS

The material presented in this report indicated that anti-jackknife devices about the fifth wheel generally produced favourable results when used to inhibit tractor jackknife. However, when the same devices were applied to trailer swing conditions, the results were usually negligible or worse than if no device had been used. This, combined with the facts that:

- jackknife produced lower torque values about the fifth wheel than trailer swing,
- trailer swing tends to regain stability on its own, and
- jackknife is attributed to a larger portion of fatal accidents than trailer swing,

leads to the conclusion that efforts should be primarily concentrated at attempting to control tractor jackknife.

Further work should be carried out to study the characteristics of jackknife and trailer swing under realistic braking conditions other than complete locking of the tires. Although lock-up represents a worst case condition, the effectiveness of anti-jackknife devices under realistic braking conditions should be examined to determine what impact braking techniques have on AJS performance.

As was discussed in the report there are, in the most general of terms, only two types of anti-jackknife devices: active and passive. Further research should be conducted to expand upon the work completed in this study. More sophisticated devices should be developed and modelled. The implementation of an active type of device requires the ability to sense the onset of tractor jackknife and distinguish it from trailer swing. Sensing, is an important area in which to explore. It should be determined whether a method of sensing can be developed that will detect jackknife (or trailer swing) for any possible configuration on the road today, under any condition. Then, an active anti-jackknife device should be developed that redistributes, to the trailer, some increment of tractor angular momentum associated with the incipient jackknife. More work should also be carried out in the area of passive viscous damping devices. The damping device showed some merit in this study and may prove to have a lesser impact on driver controllability than the other devices

A final and important area requiring further investigation concerns the effects of anti-jackknife devices on vehicle controllability and stability. The approach taken in this study was mainly a subjective analysis based on the outcome of specific simulation runs. A preferred approach to the controllability problem may be similar to the project endeavoured by the University of Toronto Institute for Aerospace Studies (UTIAS); "A Computer Model of Driver Response"(18), where a computer model simulates human response based on certain conditions. With this type of feedback, usefulness of anti-jackknife devices can be more readily determined.

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APPENDICES

APPENDIX "A"
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Whelan Motors (Truck Supplies)
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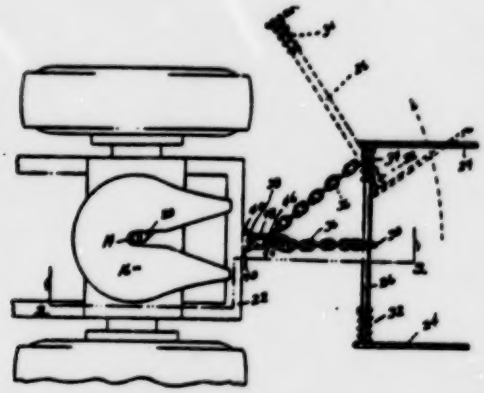
APPENDIX "B"

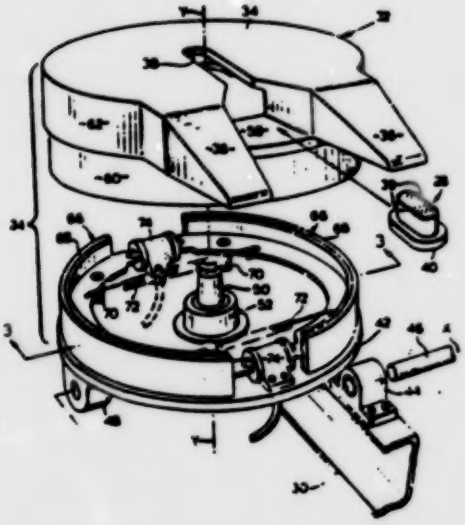
Classification of Patents

ANTI-JACKKNIFING DEVICES

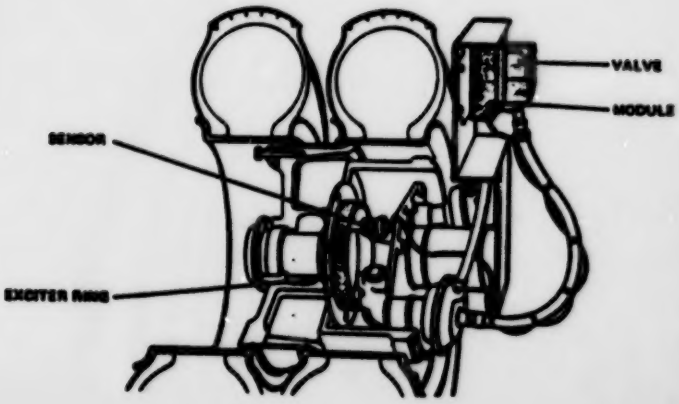
[1] JACKKNIFE CONTROL INDEPENDENT OF FIFTH WHEEL



[2] JACKKNIFE CONTROL ABOUT THE FIFTH WHEEL



[3] OTHER DEVICES (i.e. ABS)



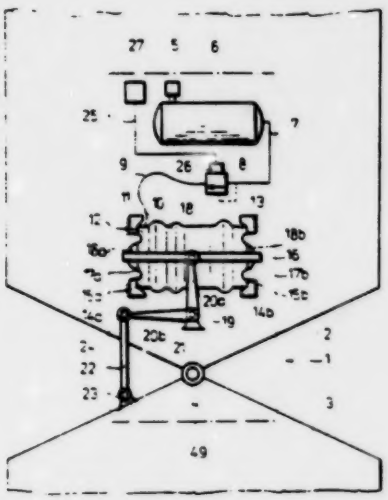
ANTI-JACKKNIFE PATENTS

[1] JACKKNIFE CONTROL INDEPENDENT OF FIFTH WHEEL

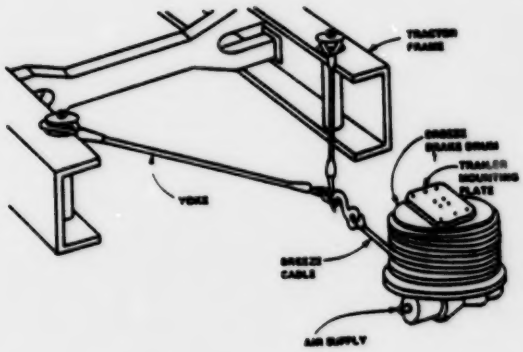
	PATENT NO.	INVENTOR
Dampening of Articulation	4 402 522 (US)	I. Ratsko
	4 366 966 (US)	I. Ratsko
	4 744 581 (US)	G. Cables
	BREEZE DEVICE	
Limiting of Articulation Angle	548 584 (CAN)	V. Bishop
	4 346 910 (US)	M. Beck

[1] JACKKNIFE CONTROL INDEPENDENT OF THE FIFTH WHEEL

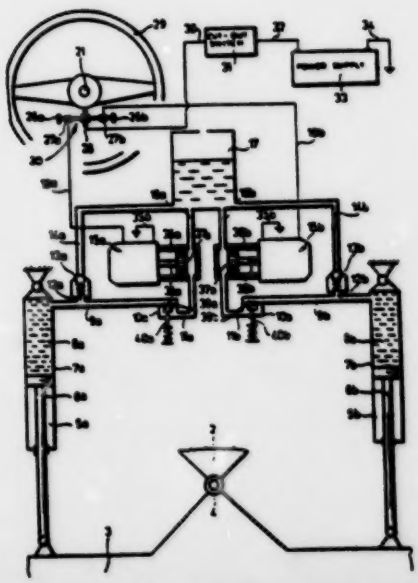
DAMPENING OF ARTICULATION



- air spring
- the larger the articulation angle, the larger the reactive force



- BREEZE type device
- brake pressure determines the amount of restraining tension in the cables

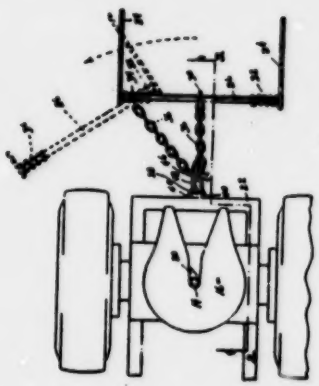


- vehicle articulation foces hydraulic fluid from the cylinders through a pressure regulating valve

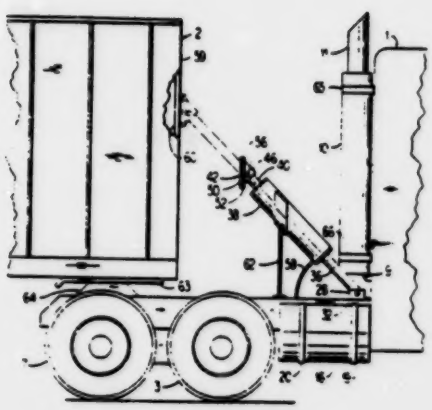
[1] JACKKNIFE CONTROL INDEPENDENT OF THE FIFTH WHEEL

LIMITING OF ARTICULATION ANGLE

- chain limits articulation angle



- hydraulic cylinders are extended when a large break pressure is reached
- assumed to be a panic situation



ANTI-JACKKNIFE PATENTS

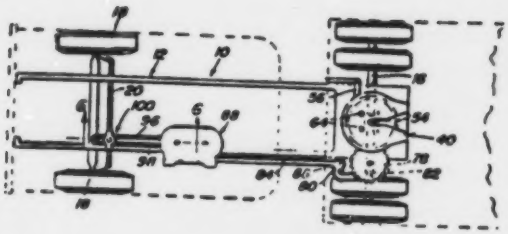
[2] JACKKNIFE CONTROL ABOUT THE FIFTH WHEEL

	PATENT NO.	INVENTOR
Control of Articulation	802 074 (CAN)	G. Neal
Dampening of Articulation	4 455 035 (US)	J. Ivony
	4 583 756 (US)	W. Hornung
	904 373 (CAN)	R. Adams
	3 580 610 (US)	
	815 414 (CAN)	F. Hope
	797 707 (CAN)	
	1 614 908 (CAN)	P. Mackey
	1 034 618 (CAN)	
	1 004 247 (CAN)	B. Roth
	4 065 149 (US)	
	MATHER DEVICE	
Limiting of Articulation Angle	4 585 248 (US)	R. Miller
	4 775 165 (US)	O. Grovum
	1 087 271 (CAN)	W. Meyers
	546 504 (CAN)	W. Safko
	661 640 (CAN)	W. Lavelle
	SAFE-T-KING	
Locking of Articulation	1 034 618 (CAN)	B. Roth

[2] JACKKNIFE CONTROL ABOUT THE FIFTH WHEEL

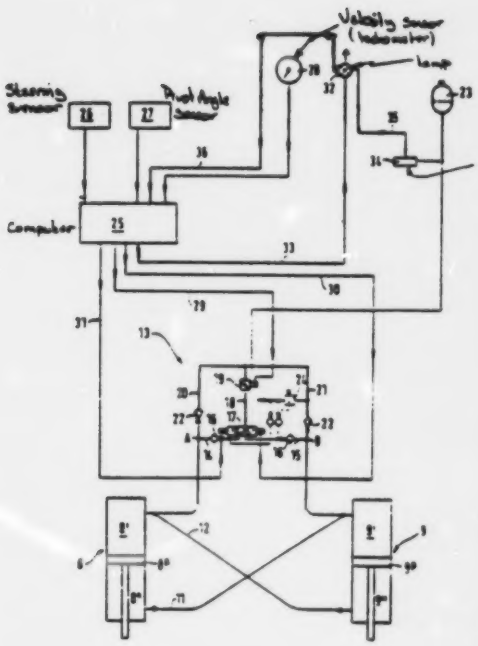
CONTROL OF ARTICULATION

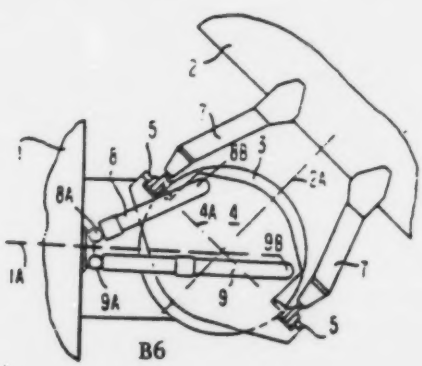
- steering is indirectly geared to the fifth wheel
- a steering input requires vehicle articulation



DAMPENING OF ARTICULATION

- hydraulic cylinders are coupled to the fifth wheel
- during articulation fluid is forced from the cylinders through a directional control valve and throttle valve
- valve action is computer controlled based on velocity, steering angle, and pivot angle inputs

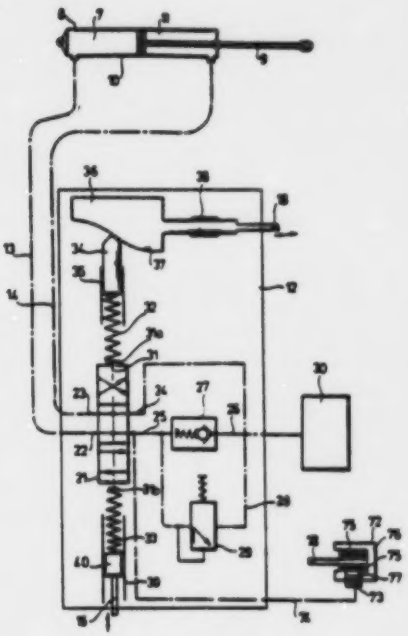




[2] JACKKNIFE CONTROL ABOUT THE FIFTH WHEEL

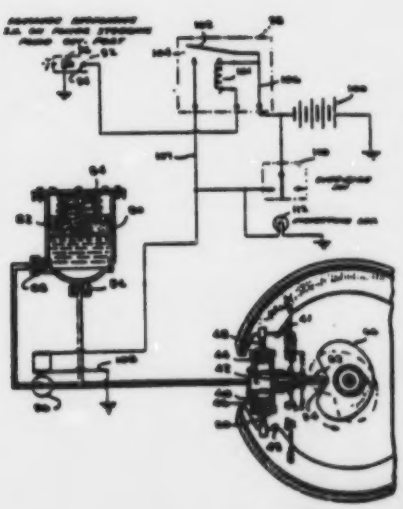
DAMPENING OF ARTICULATION (continued)

- hydraulic friction brake on fifth wheel
- activated when fluid flows through a directional control valve due to cylinder compression or extension
- valve position is determined by steering and pivot angle inputs



MATHER JACKKNIFE CONTROL UNIT

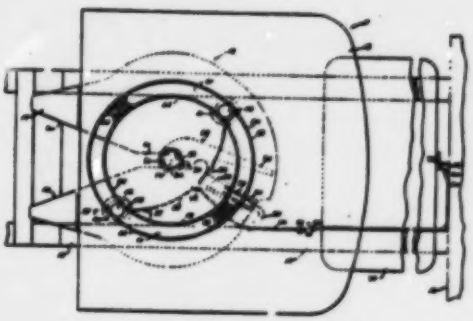
- pressurized viscous fluid is forced through a gap in a vane fixed to the king pin
- torque is proportional to the square of the articulation rate



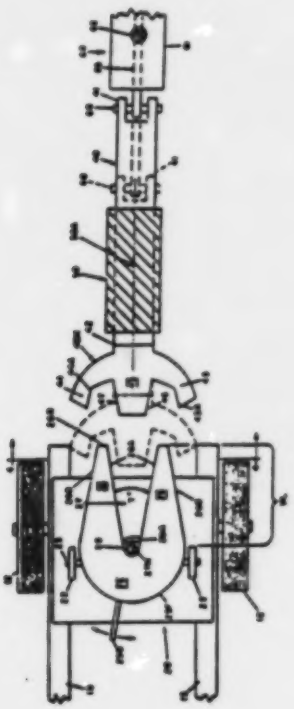
- automatic braking of the fifth wheel based on articulation angle
- cam profile determines braking force
- other activation methods: brake system, electromagnets

[2] JACKKNIFE CONTROL ABOUT THE FIFTH WHEEL

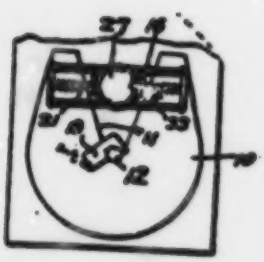
LIMITING OF ARTICULATION ANGLE



- retractable pins slide in a defined guide slot to limit articulation angle
- manually activated



- grooves in ram limit fifth wheel rotation
- manually activated

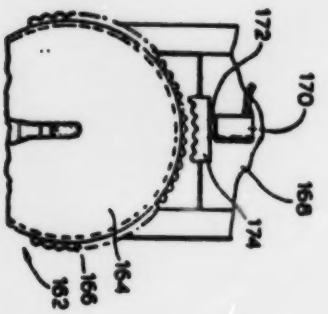


- pin slides in V-groove of fifth wheel
- springs provide some dampening before the limit is reached

[2] JACKKNIFE CONTROL ABOUT THE FIFTH WHEEL

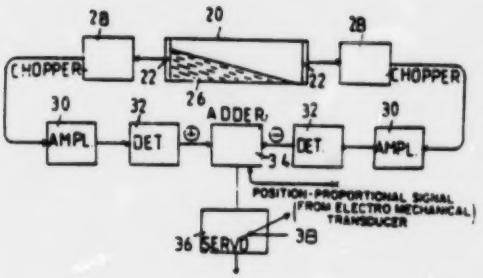
LOCKING OF ROTATION ABOUT KING PIN

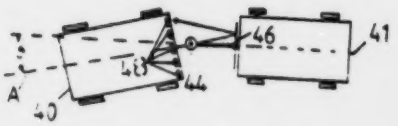
- toothed ram forced against gear fixed to rotating fifth wheel
- activated by applying brakes



[3] OTHER DEVICES

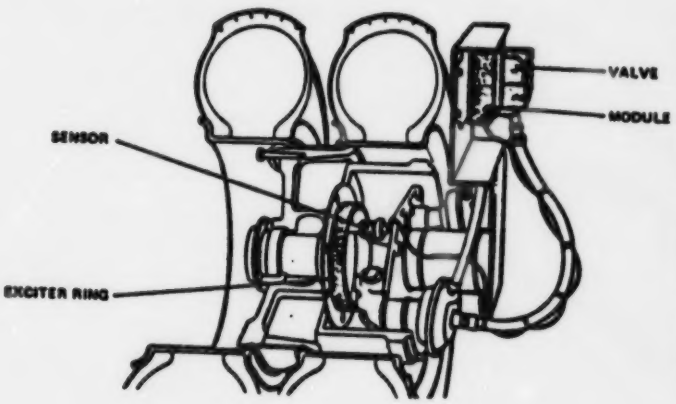
- servo control of steering, acceleration and braking based on centrifugal force sensing
- centrifugal force is detected by pressure sensitive elements or calculated based on velocity and articulation angle sensors [F=(v2)/r]

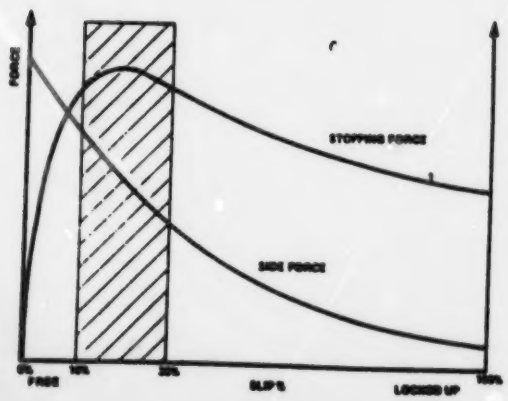


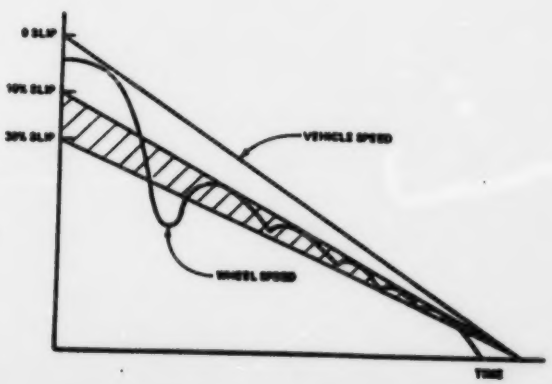


ANTI-LOCK BRAKE SYSTEMS (ABS)

- based on the ideal braking cycle of 10% to 30% slip
- ideally the wheels never slip







APPENDIX "C"

Equations of Motion

Appendix "C" Equations of Motion*

The equations of motion for the tractor semi-trailer vehicle were formulated by considering each rigid body separately and writing the equations of motion for each in terms of its yaw velocity and the longitudinal and lateral velocities of its centres of mass. By eliminating the lateral and longitudinal forces at the fifth wheel and relating the variables of motion at the fifth wheel, equations [1] through [7] are obtained describing the motion of the tractor semi-trailer on a flat surface:

$$m_t (U - Vr) + \Phi_1 = - (F_{y1} + F_{y2}) \sin \delta - (F_{y5} + F_{y6}) \sin \Psi + (F_{x1} + F_{x2}) \cos \delta + (F_{x3} + F_{x4}) + (F_{x5} + F_{x6}) \cos \Psi \quad [1]$$

$$m_s (V + Ur) + \Phi_2 = (F_{y1} + F_{y2}) \cos \delta + (F_{y3} + F_{y4}) + (F_{y5} + F_{y6}) \cos \Psi + (F_{x1} + F_{x2}) \sin \delta + (F_{x5} + F_{x6}) \sin \Psi \quad [2]$$

$$I_t r - (A2 - BB) \Phi_2 = A1 (F_{y1} + F_{y2}) \cos \delta - A2 (F_{y3} + F_{y4}) - (A2 - BB) (F_{y5} + F_{y6}) \cos \Psi + A1 (F_{x1} + F_{x2}) \sin \delta - (A2 - BB) (F_{x5} + F_{x6}) \sin \Psi + T + TRA1 (F_{x1} - F_{x2}) \cos \delta + TRA2 (F_{x3} - F_{x4}) - TRA1 (F_{y1} - F_{y2}) \sin \delta \quad [3]$$

$$I_s (r + \Psi) - A3 \Phi_3 = - (A3 + A4) (F_{y5} + F_{y6}) + TRA3 (F_{x5} - F_{x6}) - T \quad [4]$$

$$\Phi_1 = m_s (U - Vr + (A2 - BB) r^2) + A3 m_g (r + \Psi)^2 \cos \Psi + A3 m_s (r + \Psi) \sin \Psi \quad [5]$$

$$\Phi_2 = m_s (V - Ur - (A2 - BB) r) + A3 m_s (r + \Psi)^2 \sin \Psi - A3 m_s (r + \Psi) \cos \Psi \quad [6]$$

$$\Phi_3 = \Phi_2 \cos \Psi - \Phi_1 \sin \Psi \quad [7]$$

*Taken from Leucht, P.M., The Directional Dynamics of the Commercial Tractor - Semitrailer Vehicle During Braking, Society of Automotive Engineers, 1971, SAE 700371.

APPENDIX "D"

The Computer Simulation Program

APPENDIX "D" THE COMPUTER SIMULATION PROGRAM

Description of the Simulation Model

The computer simulation model chosen to conduct the analysis was entitled "The HSRI/MTC Simulation Program for a Tractor-Trailer in the Horizontal Plane". This model was based on a doctoral dissertation by Leucht entitled "Directional Dynamics of the Tractor-Semitrailer Vehicle", The University of Michigan, 1970, and was presented in an SAE paper (1).

Leucht's equations of motion were formulated into computer code by Moncarz, Bernard and Fancher of the Highway Safety Research Institute (HSRI) in 1975 (2). The resulting "TBS Simulation" program was later modified by Bernard and Kim in 1977 (3) to incorporate the Breeze jackknife restraint device. Later in 1978, the Ministry of Transportation and Communications of Ontario, MTC, (now MTO) again modified the TBS program to incorporate three other anti-jackknife devices, namely the Safe-T-King (Canada), Hope (U.K.) and Mather (U.S.A.), and an antilock braking system (4). MTC went one step further to fully validate their simulation results with experimental test data (5). This MTC model, was used in this study.

M. El-Gindy et. al. conducted a comparative study of the directional responses of commercial articulated vehicles in steady-state and lane change manoeuvres for four simulation models (6), one of which was the TBS Model. The other three models were 1) The Linear Yaw Plane Model, 2) The Yaw/Roll Model and 3) The Phase IV Model. He noted that all four models, in general, significantly deviated from measured data for lateral accelerations above 0.2 g. He also pointed out that, for the two vehicle configurations modelled in his study, there were no significant differences in the predicted steady-state steering responses for lateral acceleration up to approximately 0.25 g. However, the Phase IV, TBS and Yaw/Roll models took into account the effects of load transfer and the non-linear characteristics of the tires, and therefore predicted changes in handling behaviour with increased lateral acceleration. The Linear Yaw Plan Model did not.

Mathematical Model

The mathematical model used by Leucht (1) to represent the tractor-semitrailer vehicle is shown in Figure D1. It consisted of two rigid bodies which translate parallel to a smooth level surface and rotate about an axis perpendicular to this surface. Hence, the model incorporated four degrees of freedom (4 D.O.F.): translation of the tractor in the X and Y directions (translation of the semi-trailer is constrained by the fifth wheel coupling), rotation of the tractor in the yaw plane, and rotation of the semitrailer in the yaw plane.

A number of assumptions were made in developing the model, namely:

- rolling and pitching of the bodies on their suspensions were neglected (hence planar motion),

- each unit of the articulated vehicle was treated as a rigid body, and the unsprung masses were assumed to be rigidly attached to their respective sprung masses,
- dual tires had frictional shear forces which were twice the forces generated by a single tire operating at one half the vertical load on dual tires,
- rotational inertia of the wheels was neglected, and
- aerodynamic forces and moments were neglected.

As was previously discussed, load transfer due to motion of the vehicle was accounted for in this model. These load transfers were not determined as a function of the displacement and velocity of the suspension components, but rather as a quasi-static function of the acceleration of the centre of gravity of each mass element. This method allowed for the contribution of the pitch and roll of the mass elements without treating the motions as degrees of freedom.

All external forces acting on the vehicle were the normal and friction forces applied through the tire-road interface. The frictional shear force of the tire can be resolved into two components in the plane of motion as shown in Figure D1: F_y, the lateral force perpendicular to the plane of wheel rotation, and F_x the longitudinal shear force in the direction of wheel heading. The normal force had a large influence on the magnitude of these shear forces and was determined by the sum of the static load and any instantaneous load transfer resulting from pitch and roll moments due to vehicle manoeuvres (as was discussed above).

Tire Model

Modelling at the tire road interface was a very important aspect of the simulation as it represented all external forces acting on the vehicle. The model coded by Moncarz et al (2), in their TBS simulation, was based on a formulation first offered by Fiala. This differed from the model proposed by Leucht (1), which was based on the lateral force characteristics defined by Nordeen. The difference stemmed from the way in which the lateral side forces of the tires were treated. Leucht looked at the lateral forces as differing under two conditions. The wheels were assumed free rolling when the longitudinal braking force on the tire remained less than the longitudinal component of the friction force, given by the following inequality (refer to nomenclature at front of report):

$$- FS_{xi} < MUP\ F_{zi} \cos \alpha_i \qquad [1]$$

When the inequality was violated, the wheel was assumed locked and a coulomb friction process existed, yielding the following relationships for F_{xi} and F_{yi}.

$$F_{xi} = - MUP\ F_{zi} \cos \alpha_i \qquad [2]$$

$$F_{yi} = - MUP\ F_{zi} \sin \alpha_i \qquad [3]$$

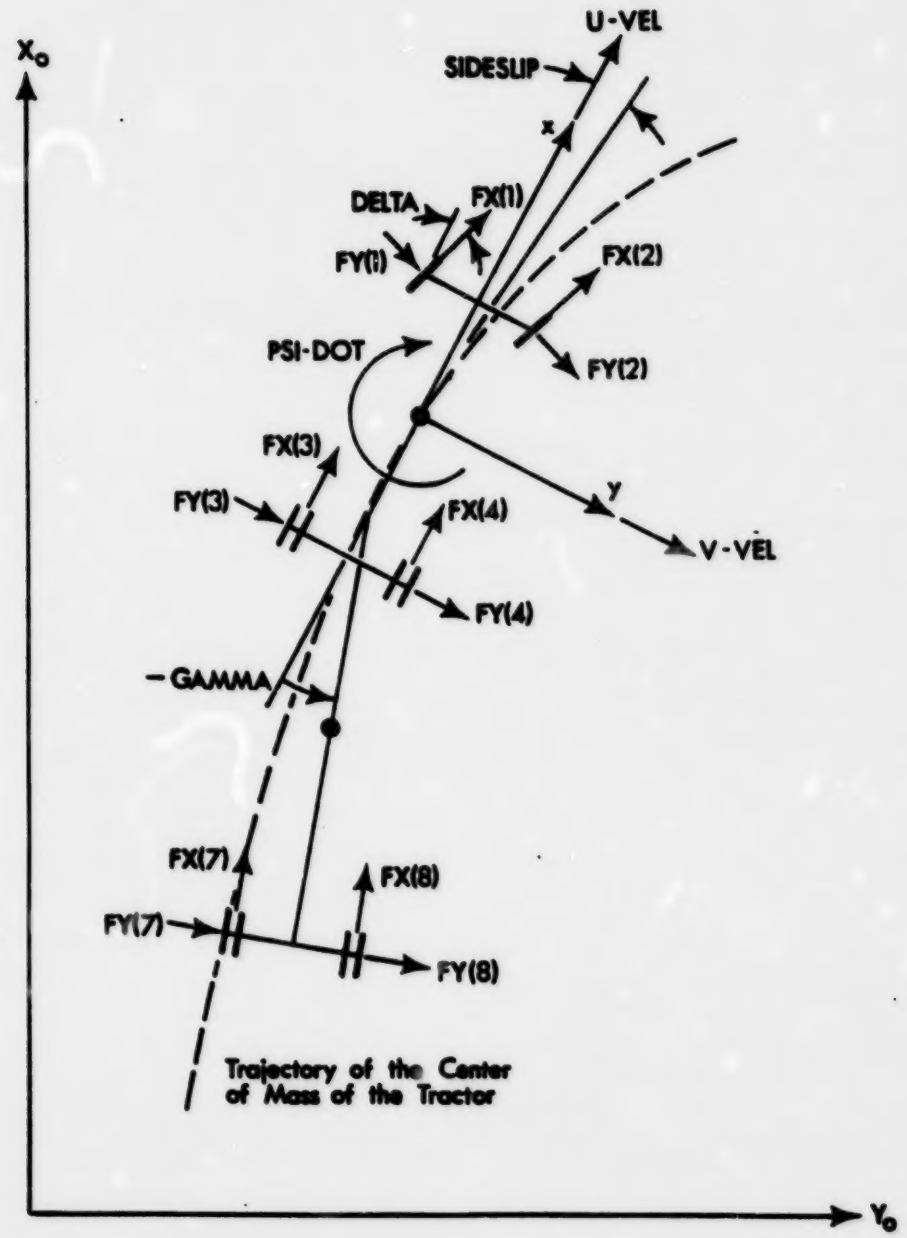


Figure D1 Tractor-Semitrailer Vehicle Model

Otherwise the forces on the assumed free rolling tire were given by:

$$FS_{xi} = F_{xi} \qquad [4]$$

$$F_{yi} = F_{zi} \cdot f(\alpha_i, \bar{F}_{zi}) \qquad [5]$$

Figure D2 shows the non linear relationship between the lateral force, normal load and slip angle for the free rolling tire.

Moncarz et al (2) took the approach of looking at the shear forces at the tire road interface in three distinct categories, namely, lateral forces of the free rolling tire, longitudinal forces of the tire in straight line braking, and tire forces in combined braking and steering. This topic was presented in references (2) and (4), but is presented again for clarification and completeness.

Lateral Forces of the Free Rolling Tire

The lateral force characteristics of a free rolling tire are a nonlinear function of the slip angle (the angle between the direction of wheel travel and the direction of wheel heading) and the normal load. As quoted in (2), Fiala formulated the lateral force, F_{yi}, of the free rolling tire as:

$$F_{yi} = - MUP \cdot F_{zi} \left(\sigma - \frac{\sigma|\sigma|}{3} + \frac{\sigma^3}{27}\right) \qquad \text{for } \sigma < 3$$

$$F_{yi} = - MUP \cdot F_{zi}\, \sigma\, \frac{\sigma}{|\sigma|} \qquad \text{for } \sigma > 3 \qquad [6]$$

where

$$\sigma = \frac{CALF \cdot \alpha}{MUP \cdot F_{zi}}$$

Careful choice of CALF and MUP results in a reasonable fit to measured tire data. Care must be taken in choosing values for CALF and MUP as they are load and speed sensitive and could deviate from real values under severe load transfer situations.

Longitudinal Forces of the Tire in Straight-Line Braking

This aspect of the tire braking model was very straight forward. An attempted brake force was determined based on the user specified brake force table. If this brake force FS_{xi} was less than $MUP \cdot F_{zi}$, then the simulated brake force, F_{xi}, was set equal to the attempted brake force. Otherwise, the wheel was considered locked and the simulated brake force became equal to $-MUS \cdot F_{zi}$.

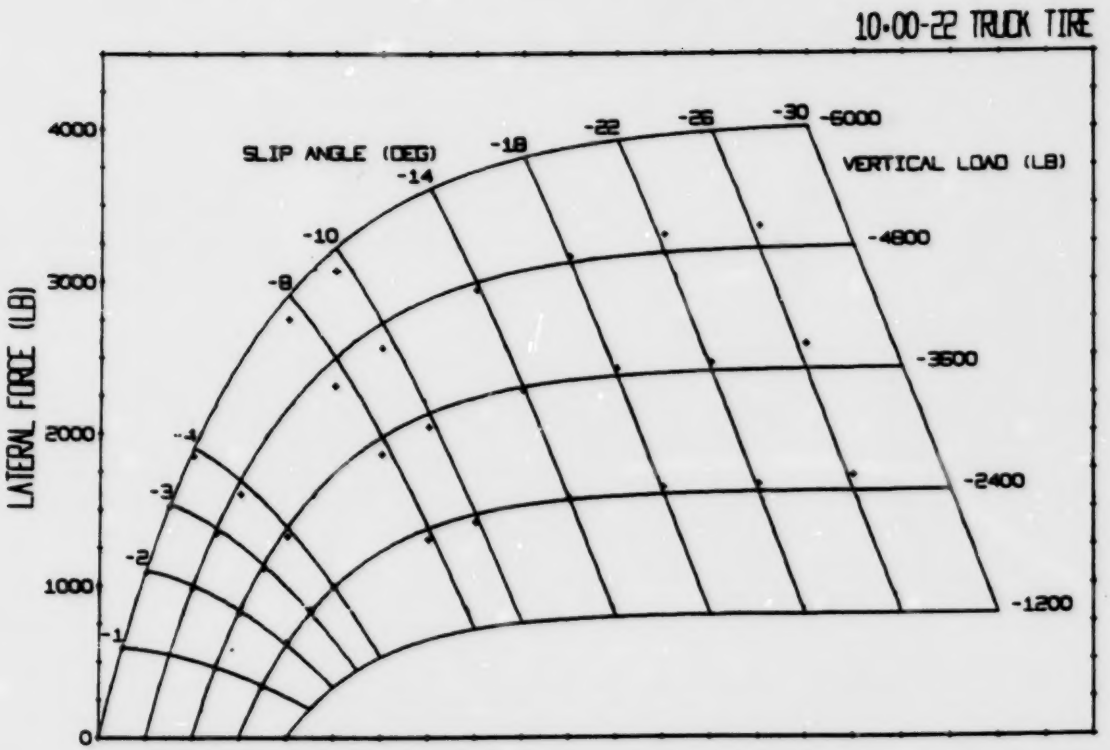


Figure D2 Tire Lateral Force Characters

Tire Shear Forces Due to Combined Steering and Braking

Manoeuvres involving combined steering and braking resulted in tire shear forces in both the longitudinal and lateral directions.

The longitudinal friction coefficient decreased as the slip angle, α increased. This modified friction coefficient was determined by

$$MUMOD = MUP\ (1.0 - 1.7\ \alpha) \qquad [7]$$

Thus

$$F_{xi} = MUMOD \cdot F_{zi} \qquad [8]$$

if lockup occurred, the longitudinal component of the shear force was given by

$$F_{xi} = MUS \cdot F_{zi} \cos(\alpha) \qquad [9]$$

The magnitude of the lateral shear force is a function of tire slip. Since the program neglected to treat the wheel spin as a degree of freedom, the spin dynamics were accounted for using the quasi-static calculations.

The user is required to input three parameters MUP, the peak friction coefficient, SP, the value of longitudinal slip at which MUP occurs, and MUS, the friction coefficient of the sliding or locked wheel. These values were used to form the three point MU-slip curve shown in Figure D3. From this curve the longitudinal slip was estimated by

$$S = -\ SP \cdot \frac{F_{xi} / F_{zi}}{MUP} \qquad \text{if } | FS_x | \leq MUP \cdot F_{zi}$$

$$= 1.0 \qquad \text{otherwise (locked wheel)} \qquad [10]$$

For a known value of S at an instant in time, the lateral force on the tire was determined by:

$$F_{yi} = F_{yi}^{*} \cdot F(S) \qquad [11]$$

where F_{yi} was the lateral force due to combined braking and steering, F_{yi}^{*}, was the lateral force of the free rolling tire, and F(S) was the "slip roll off function". The function F(S) is empirical and was imbedded into the computer code, where it acted as a weighting factor for varied values of slip, S. Figure D4 depicts the slip roll off function, F(S), used in the simulation. It should be noted that these values can be changed by the user if so desired.

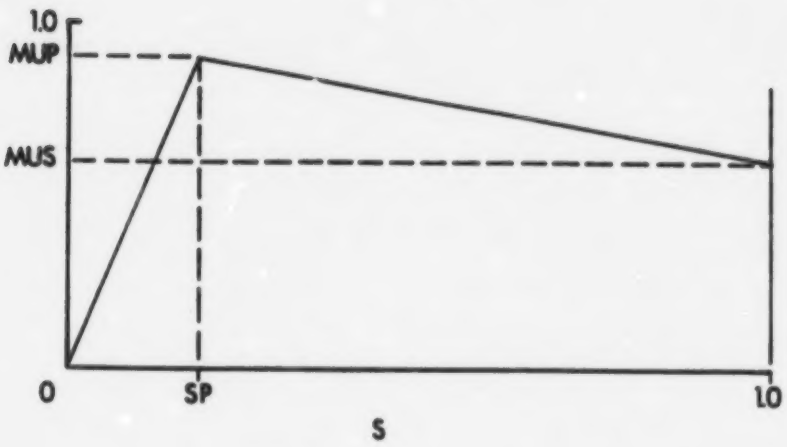


Figure D3 MU-Slip Curve

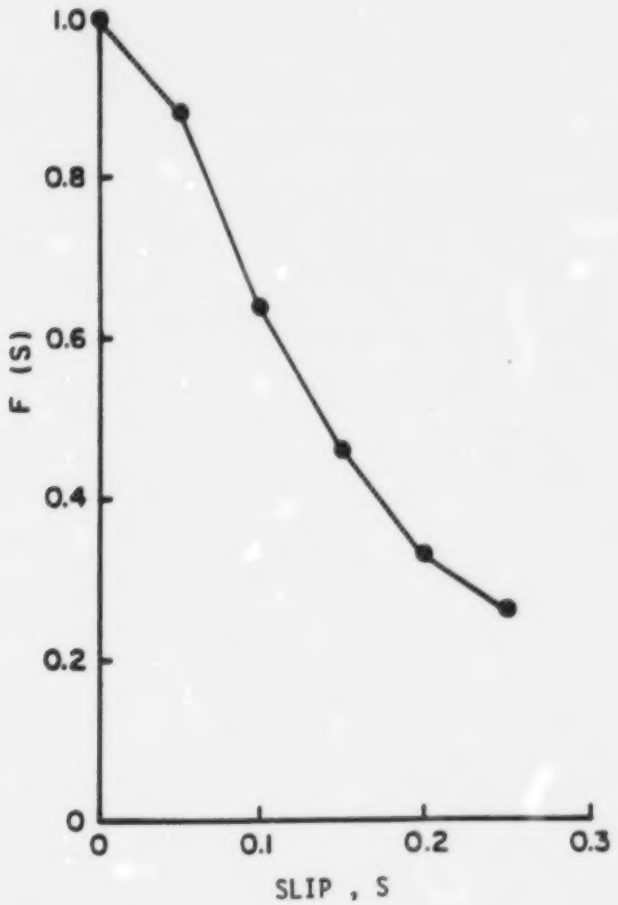


Figure D4 Slip Roll Off Function, F(s)

In the locked tire case the friction force at the tire road interface must oppose the direction of motion. Equation [9] established the longitudinal force due to the locked tire, thus, the lateral force must be

$$F_{yi} = MUS \cdot F_{zi} \cdot \sin(\alpha) \qquad [12]$$

Braking and Steering

The combined steering and braking capability of the program was one of the major reasons for its selection. Both braking and steering were handled by specifying inputs in tabular format.

Steering

The average steer angle of the front wheels (left and right were assumed equal) was entered as an ordered pair with time. For example a steer table with the lines:

Time	Steer
0.0	0
0.5	3.0
1.0	3.2

would produce the steer input depicted in Figure D5.

Up to twenty-five lines could be entered. During execution of the program, if computations were required at time t which falls between two specified points, the steer angle at that time was determined by linear interpolation between the two points. If t was larger than the last time entered, the steer angle was set equal to the last value entered.

Braking

Values for attempted brake application were entered in a similar manner to the steer inputs. In this case, the time history for the brake application on each side of an axle was entered on a single line. With reference to Figure D6, brake forces were entered for tires 1, 2, 3, 4, 7 and 8. Dual tires were treated effectively the same as single tires, and the trailing axle of a tandem was assigned the same value as the leading axle. As an example, the lines

Time	Tire 1	Tire 2	Tire 3	Tire 4	Tire 6	Tire 7
0	0	0	0	0	0	0
.5	0	0	0	0	0	0
1.5	20,000	20,000	25,000	25,000	0	0
2.0	20,000	20,000	25,000	25,000	0	0
3.0	20,000	20,000	25,000	25,000	35,000	35,000

would produce a brake input to the program as shown in Figure D7. As may be evident from the above description, it was possible to simulate brake imbalance on a single, or tandem axle.

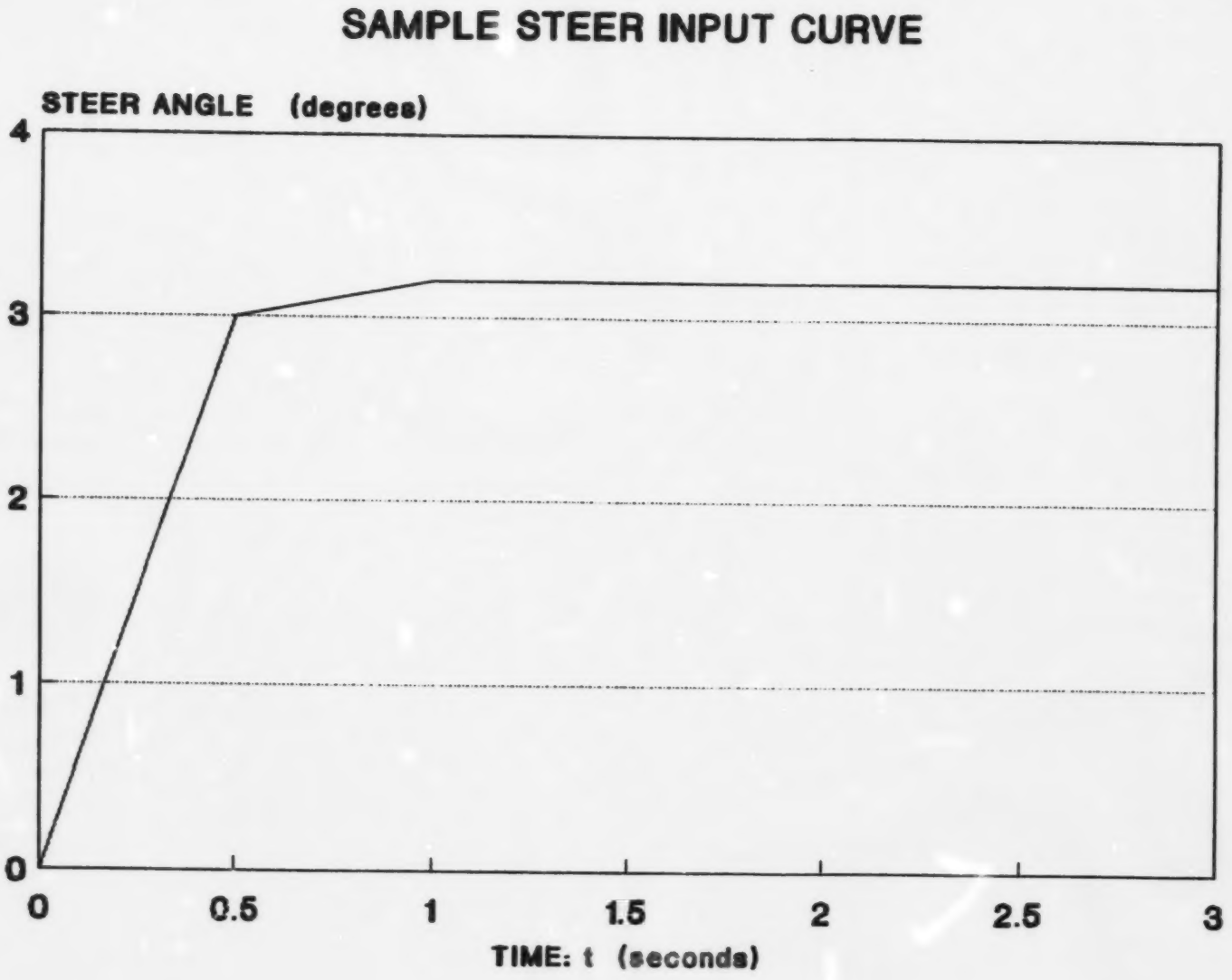


Figure D5 Sample Steer Input Curve

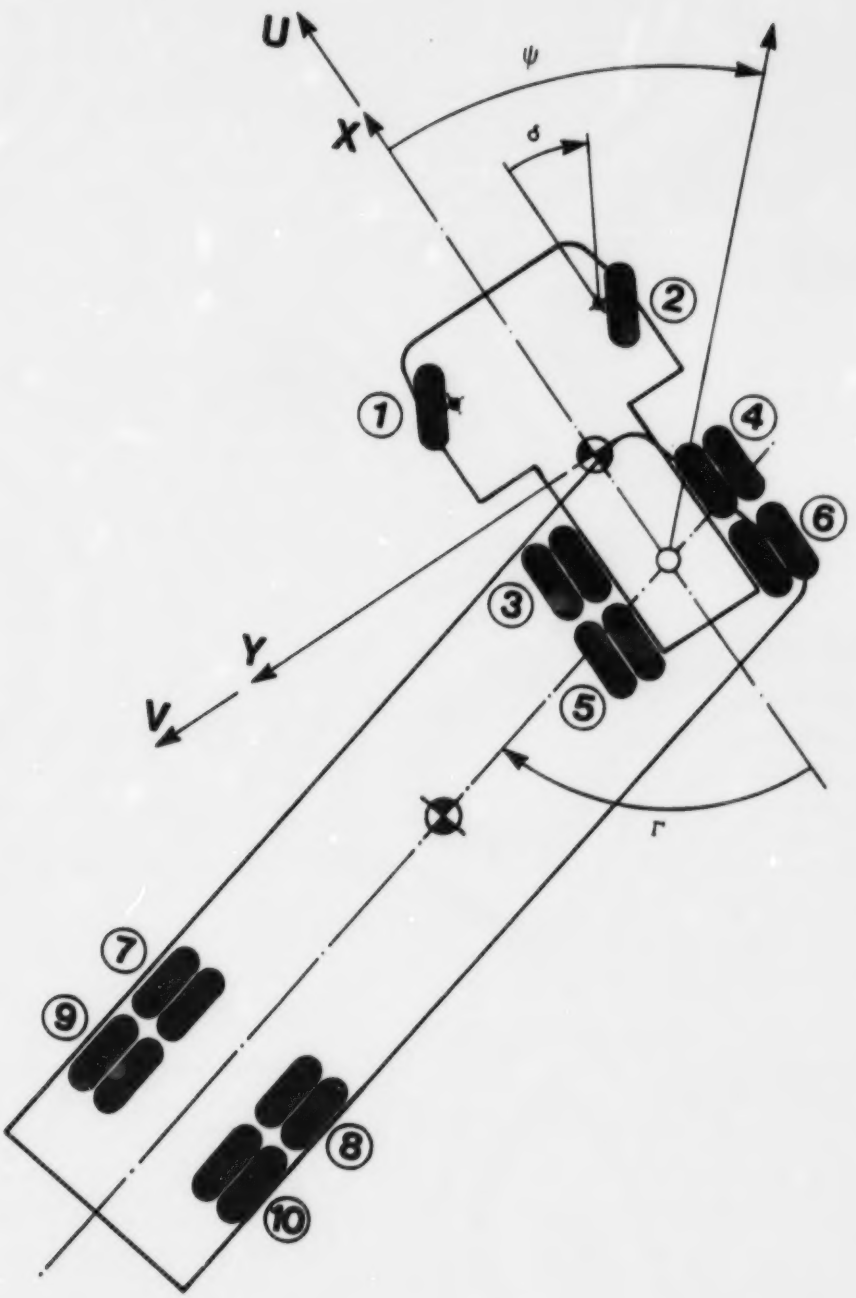


Figure D6 Tire Numbering Convention

SAMPLE BRAKE INPUT CURVE

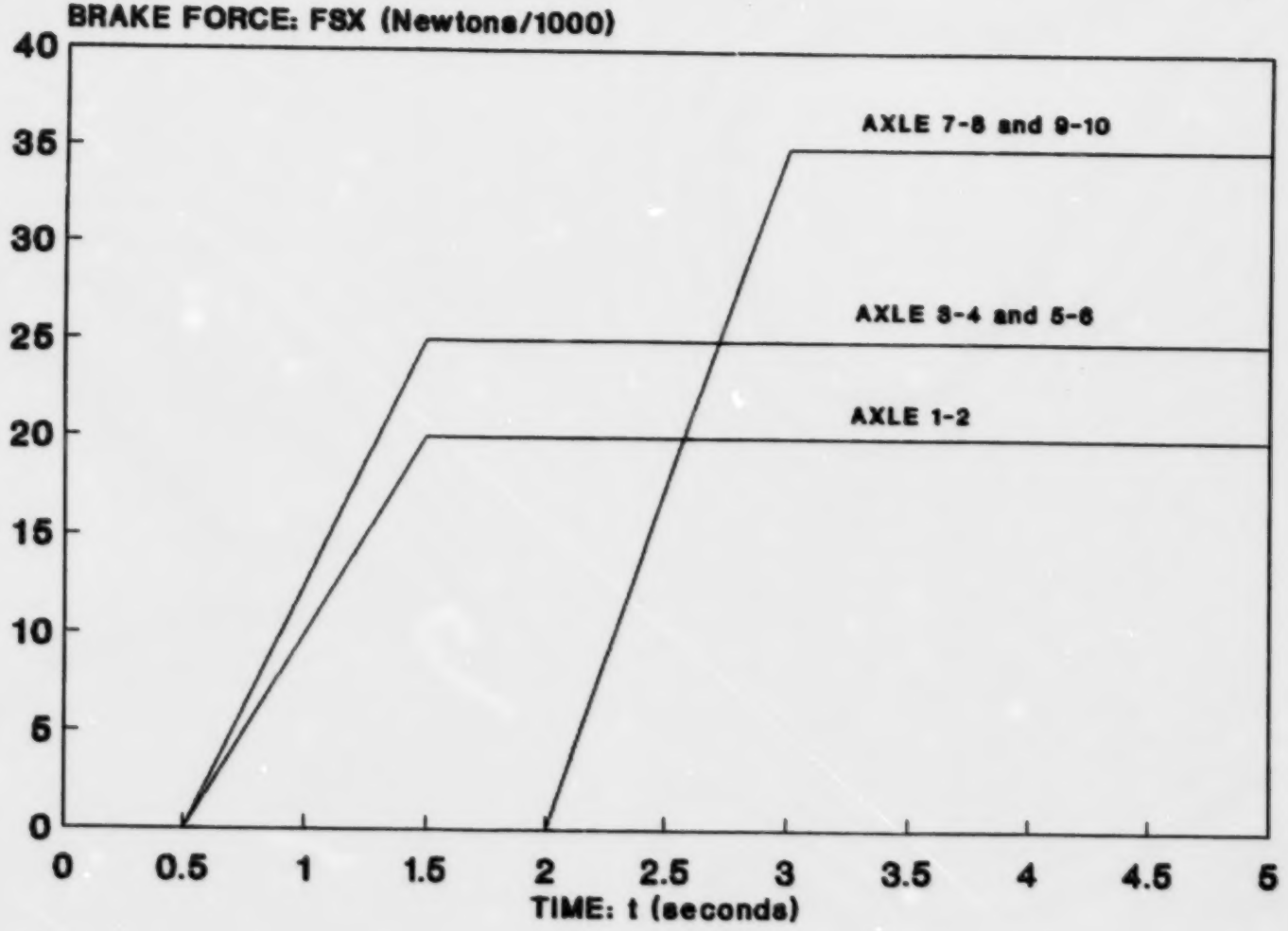


Figure D7 Sample Brake Input Curve

Program Inputs

The program requires the input of 40 lines of data, excluding the brake and steer tables, in order to successfully execute. These values may be entered either interactively or in the form of an input file as shown in Figure D8.

Line 1 is a three digit code used to assign anti-lock brakes to the axles of the configuration. A "0" indicates that ABS is not active and a "1" indicates that it is. The first digit represents the tractor front axle, axle 1-2, (see Figure D1), the second digit applies to the tractor rear axle, axle 3-4, and the third to the semitrailer axle, axle 7-8. Similar to the brake force table, the value entered for the leading axle of a tandem pair, applies to the trailing axle. Therefore axle 5-6 will assume the same anti-lock code as axle 3-4, and axle 9-10 will be equivalent to axle 7-8.

Line 2 is the four digit code used to determine if the vehicle components are modelled with tandem axles and/or dual tires. As with "Line 1" a "0" indicates no and a "1", yes. A value of "1" as the first digit specifies a tractor rear tandem axle (a "0" would indicate a single axle). The second digit indicates if the tractor rear axle(s) is to have dual or single tires, a "1" indicates dual. Similarly, the final two digits apply to the semitrailer axle(s).

Line 3 and Line 4, as indicated in Figure D1, are the gross vehicle weight of the tractor and semitrailer, respectively.

Line 5 and Line 6 are the yaw moments of inertia of the tractor and semitrailer respectively. These values may be approximated (7) by:

$$IZZ = ITZZ = (L/2)^2 \, GVW/386 \qquad [13]$$

Line 7 through to Line 19 are the geometric dimensions of the tractor and semitrailer and are depicted in Figure D9.

Line 20 and Line 21 show the characteristics of the fifth wheel where Line 20, MU5, is the fifth wheel friction coefficient, and Line 21 is the effective radius of the fifth wheel. These variables are used to calculate the coulomb friction about the fifth wheel.

Line 22, GAM1, is the fraction of the total lateral load transfer that takes place at the front axle of the tractor. This can be estimated by dividing the front roll moment per unit of roll angle for a steady turn by the sum of the front and rear roll moments.

Line 23, GAM3, is used to calculate the quasi-static inter-axle load transfer. The product of this coefficient and the brake force on one side of the tandem axles gives the inter-axle load transfer for that side (16). For a four spring suspension, GAM3 (or GAM4) can be approximated with a value of -0.5. For a walking beam suspension, GAM3 and GAM4 can be approximated by

$$(AA/R_t) \, (100 - TE) \, / \, 100 \qquad [14]$$

```
000                 41;anti-lock code: trac front. rear. trlr (0=n.1=y)
1111                42;tandem code: trac rear. dual. trlr. dual (0=n.1=y)
9047.               01.GVW1;wt. of trac. (kg)
2948.               02.GVW2;wt. of trailer. (kg)
62000.              03.IZZ;mom. of inert. of trac. (kg-m*m)
46927.              04.ITZZ;mom. of inert. of trailer. (kg-m*m)
1.52                05.AA;dist. between tractor tandem axles (m)
1.22                06.AAT;dist. between trailer tandem axles (m)
2.565               07.A1;dist. from tractor cg to front axle (m)
3.025               08.A2;dist. from tractor cg to rear axle (m)
3.94                09.A3;dist. from trailer cg to 5th wheel (m)
2.46                10.A4;dist. from trailer cg to rear axle (m)
0.1524              11.BB;dist. from tractor rear suspension to 5th wheel (m)
1.029               12.TRA1;1/2 lat. dist. between tractor front tire to cl (m)
.991                13.TRA2;1/2 lat. dist. between tractor rear tire to cl (m)
1.03                14.TRA3;1/2 lat. dist. between trailer tire to cl (m)
1.32                15.Z0;ht. of 5th wheel above ground (m)
1.118               16.Z1;ht. of tractor cg above ground (m)
1.65                17.Z2;ht. of trailer cg above ground (m)
.05                 18.MU5;5th wheel friction coef.
.0483               19.???;equivalent radius of 5th wheel (m)
.110               20.GAM1;portion of lat. load trans. on front axle of tractor
0.15                21.GAM3;tractor tandem axle load trans. coef.
0.15                22.GAM4;trailer tandem axle load trans. coef.
120.                23.VEL;initial velocity (U-direction) (km/h)
15.0                24.TIMF;max. simulation time (sec.)
360.0               25.IQUIT;max. articulation angle allowed (deg)
4300.               29;calf(1) \
4300.               30;calf(2) | cornering stiffness of tires (N/deg)
4300.               31;calf(3) /
NO                  ;do all tires have the same MU-slip curve?
.47                 32;mup(1) \
.47                 33;mup(2) | peak tire-road friction coef.
.47                 34;mup(3) /
.25                 35;mus(1) \
.25                 36;mus(2) | sliding tire-road friction coef.
.25                 37;mus(3) /
.15                 38;sp(1) \
.15                 39;sp(2) | slip corresponding to peak MU.
.15                 40;sp(3) /
02                  ;no. lines in brake table. lines to follow---\/
0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0
02                  ;no lines in steer table. lines to follow---\/
0 0
1 1.0
0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0     ;AJS device. 9,parameters...
0                   ;TES/ajs enable code (0=no, 1=yes)
END OF DATA
File: Sample.inp
```

Figure D8 Sample Input File

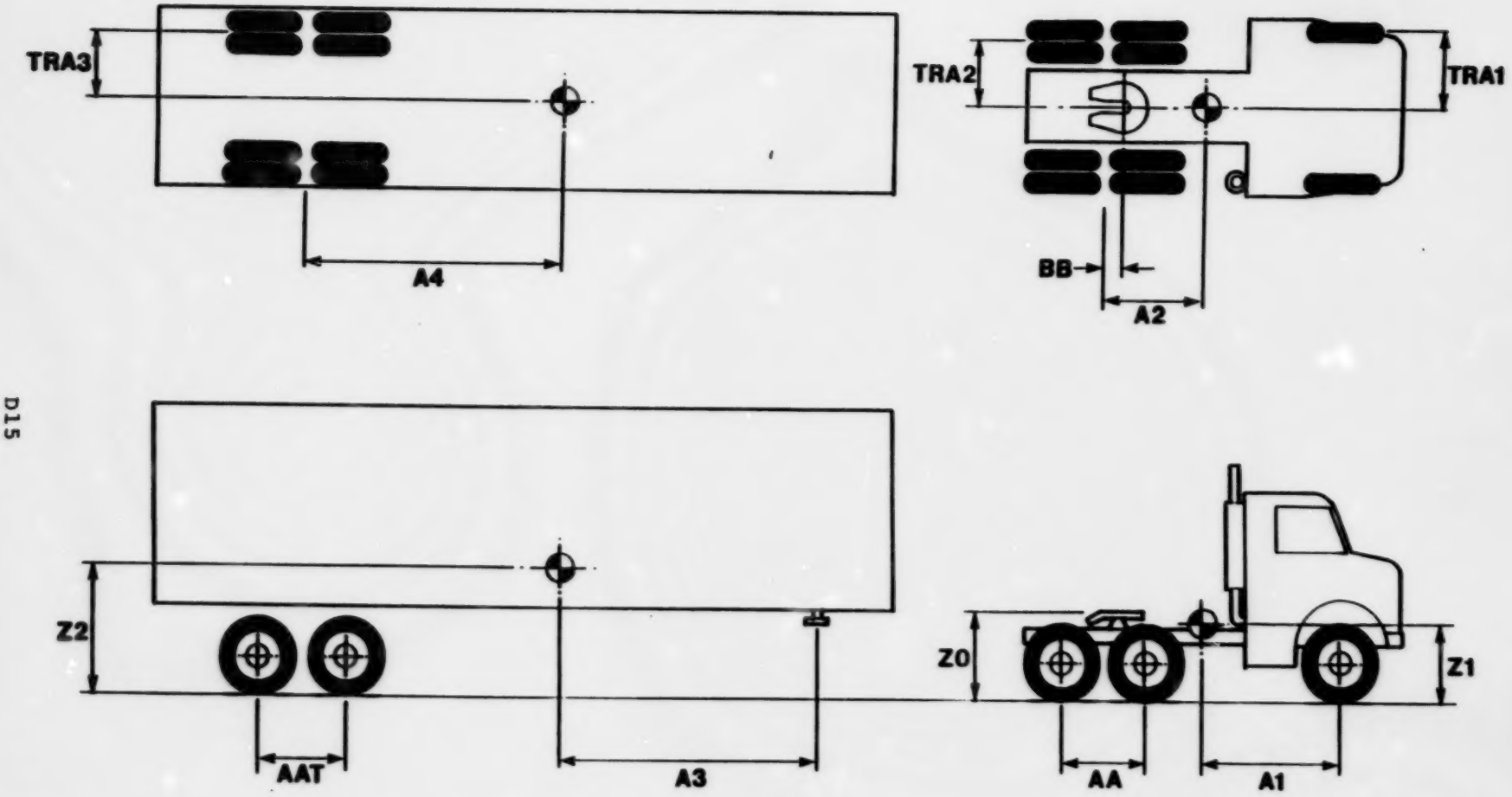


Figure D9 Vehicle Geometric Dimensions

where R_t is the radius of the tire and TE is the torque rod effectiveness. A typical value for a walking beam suspension is +0.15.

Line 24, GAM4, similar to GAM3, but with respect to the semitrailer.

Line 25, VEL, is the initial starting velocity in km/hr, if it becomes 0 or negative, simulation is stopped.

Line 26, TIMF, is the duration of the simulation exercise (stopping criteria).

Line 27, IQUIT, is the maximum articulation angle, in degrees, allowed. If it is exceeded, simulation stops.

Line 28, Line 29, and Line 30 CALF(*), are values that are entered for the cornering stiffness, in N/deg, of the tires on the axle. CALF(1) corresponds to the tractor front axle, CALF(2) to the tractor rear axle(s), CALF(3) to the trailer axle(s).

Line 31, Line 32 and Line 33, MUP(*), are values used for peak friction coefficients of the tires on the axle, Subscripts (*) correspond to the axles as described above.

Line 34, Line 35, and Line 36, MUS(*), similar to the above, are values entered for the sliding friction coefficients.

Line 37, Line 38 and Line 39, SP(*), are the values of slip corresponding to MUP for the respective axles. The values of MUP, MUS, and SP are combined to produce the MU-slip curve shown in Figure D3.

Line 40 is used to activate a particular anti-jackknife device. The first value indicates which device is to be active and the remaining nine values are parameters that are characteristic to that device. Refer to Table D-1.

Finally Line 41 is used to enable of disable the T E S anti-jackknife device. A "0" indicates that the device is disabled, and a "1" that is enabled. The value is used to route the calculation of the restraining torque about the fifth wheel to a separate subroutine.

Program Outputs

A total of 85 output variables are available to be routed to either an output file or to the terminal screen. This is accomplished by specifying the corresponding variable number, as shown in Figure D10. One minor drawback to the program is that only 24 variables may be outputted to a file and only 6 to the screen. Figure D11 shows a sample output file.

It should be noted at this point that the time increments printed out may vary as much as 0.02 seconds from the desired values. This is due to the method of integration of the HPCG subroutine, and the manner in which the next time step to be displayed is triggered (2).

Table D1 Jackknife Control Device Input Data

VARIABLE	TYPICAL VALUE	UNITS	MEANING
KAJD	0	-	No device active
PARJ(1-9)	0.0	-	Not used
KAJD	1	-	BREEZE device active
PARJ(1)	41.9	N/kPa	k_B (Equation 9)
(2)	52.0	kPa	p_e (Equation 9)
(3)	1.966	m	X_1)
(4)	.396	m	X_2) (Figure 11)
(5)	.408	m	X_3)
(6)	3.018	m	X_4)
(7)	.012	m	ΔX, Cable slack
(8)	2.5	deg	ΔY, articulation angle adjustment
(9)	1.091	m	Z_B, BREEZE cable height above ground
KAJD	2	-	HOPE device active
PARJ(1)	41.9	N/kPa	k_B (Equation 9)
(2)	52.0	kPa	p_e (Equation 9)
(3)	0.0	N.m/rad/s	Dynamic gain
(4-9)	0.0	-	Not used
KAJD	3	-	MATHER device active
PARJ(1)	440 000.	$N.m/(r/s)^2$	k_M (Equation 18)
(2-9)	0.0	-	Not used
KAJD	4	-	SAFE-T-KING device active
PARJ(1)	12.0	deg	γ_o (Figure 3)
(2)	.279	m	R^o (Figure 3)
(3)	13.0	deg	γ_1)
(4)	37800.	N	F_1)
(5)	15.3	deg	γ_2) (Figure 14)
(6)	57800.	N	F_2)
(7)	15.9	deg	γ_3)
(8)	650 000.	N	F_3)
(9)	190 000.	N.m/(rad/s)	k_s (Equation 16)
KEYANT	111	-	KELSEY-HAYES device active (all axles)
ANTEFF(1)	0.28	-	Lateral antilock effectiveness
ANTEFF(2)	0.32	-	Longitudinal antilock effectivess
KAJD	0	-	No other device active
PARJ(1-9)	0.0	-	Not used

```
*** POSITION VARIABLES ***

 1          XO-COORD
 2          YO-COORD
 3          PSI
 4          GAMMA
*** VELOCITY VARIABLES ***

 5          U-VEL
 6          V-VEL
 7          PSIDOT
 8          GAMMADOT
***

 9          TURN-RAD
10          SIDESLIP
*** TIRE SLIP ANGLES ***

11          ALFA1+2
12          ALFA3+4
13          ALFA5+6
14          ALFA7+8
15          ALFA9+10
*** ACCELERATION VARIABLES ***

16          U-DOT
17          V-DOT
18          PSI-DDOT
19          GAM-DDOT
20          LONG.ACC
21          LAT.ACC
*** TIRE-ROAD INTERFACE FORCES ***
*** BRAKE FORCES: FX(I), SIDE FORCES FY(I) ***

22          FX(1)
23          FX(2)
24          FX(3)
25          FX(4)
26          FX(5)
27          FX(6)
28          FX(7)
29          FX(8)
30          FX(9)
31          FX(10)
32          FY(1)
33          FY(2)
34          FY(3)
35          FY(4)
36          FY(5)
37          FY(6)
38          FY(7)
39          FY(8)
40          FY(9)
41          FY(10)
```

Figure D10 Output Variables (14)

```
*** LOAD TRANSFERS, LONG. DFX(I),LAT. DFY(I) ***

42              DFX(1)
43              DFX(2)
44              DFX(3)
45              DFX(4)
46              DFX(5)
47              DFX(6)
48              DFX(7)
49              DFX(8)
50              DFX(9)
51              DFX(10)
52              DFY(1)
53              DFY(2)
54              DFY(3)
55              DFY(4)
56              DFY(5)
57              DFY(6)
58              DFY(7)
59              DFY(8)
60              DFY(9)
61              DFY(10)
*** INSTANTANEOUS LOAD FORCES ***

62              FZ(1)
63              FZ(2)
64              FZ(3)
65              FZ(4)
66              FZ(5)
67              FZ(6)
68              FZ(7)
69              FZ(8)
70              FZ(9)
71              FZ(10)
*** PROGRAMMED BRAKE FORCES ***

72              FSX(1)
73              FSX(2)
74              FSX(3)
75              FSX(4)
76              FSX(5)
77              FSX(6)
78              FSX(7)
79              FSX(8)
80              FSX(9)
81              FSX(10)
*** HITCH FORCES ***

82              XHITCH
83              YHITCH
*** J-K CNTL DEVICE TORQUE ***

84              AJK-TORQ
*** STEER ANGLE ***

85              STEER
```

Figure D10 Output Variables (continued)

MTC-HSRI INTERACTIVE TRACTOR-SEMITRAILER SIMULATION PROGRAM

RUN NO. 1
Input File: Sample.inp
Date: March 27, 1989

SIMULATION PARAMETERS

INERTIAL:

GVW1 =	9047.00 KG	A1 =	2.57 M
GVW2 =	2948.00 KG	A2 =	3.03 M
IZZ =	62000. KG.M.SEC**2	A3 =	3.94 M
IZZT =	46927. KG.M.SEC**2	A4 =	2.46 M
		Z1 =	1.12 M
		Z2 =	1.65 M

GEOMETRIC:

AA =	1.52 M	TRA1 =	1.03 M
AAT =	1.22 M	TRA2 =	0.99 M
ZO =	1.32 M	TRA3 =	1.03 M
BB =	0.15 M	KEYAT =	1111

SUSPENSION & FIFTH WHEEL:

GAM1 = 0.110	MU5 = 0.05
GAM3 = 0.150	RAD5 = 0.05 M
GAM4 = 0.150	

TIRES:

MUP(1) = 0.470	CALF(1) =4300.00 N/DEG
MUP(3) = 0.470	CALF(3) =4300.00 N/DEG
MUP(7) = 0.470	CALF(7) =4300.00 N/DEG
MUS(1) = 0.250	SP(1) = 0.150
MUS(3) = 0.250	SP(3) = 0.150
MUS(7) = 0.250	SP(7) = 0.150

ANTILOCK:
KEYANT = 000

EFFECTIVENESS:

LAT(1) = 0.000	LONG(1) = 0.000
LAT(3) = 0.000	LONG(3) = 0.000
LAT(7) = 0.000	LONG(7) = 0.000

ANTIJACKKNIFE DEVICE:
KAJD = 0

PARJ(1) =	0.00	PARJ(5) =	0.00
PARJ(2) =	0.00	PARJ(6) =	0.00
PARJ(3) =	0.00	PARJ(7) =	0.00
PARJ(4) =	0.00	PARJ(8) =	0.00
		PARJ(9) =	0.00

RUN:

VEL =120.00 Km/h	IQUIT =360.00 DEG
TIMF = 15.00 SEC	

Figure D11 Sample Output File

MTC-HSRI INTERACTIVE TRACTOR-SEMITRAILER SIMULATION PROGRAM

RUN NO. 1
Input File: Sample.inp
Date: March 27, 1989

TIME	PSIDOT	GAMMA	LAT.ACC	STEER	U-VEL
0.00	0.00	0.00	0.00	0.00	33.33
0.50	0.98	-0.15	0.57	0.50	33.33
1.02	2.12	-0.44	1.20	1.00	33.33
1.52	2.22	-0.56	1.29	1.00	33.32
2.02	2.21	-0.56	1.29	1.00	33.32
2.52	2.21	-0.56	1.29	1.00	33.31
3.02	2.21	-0.56	1.29	1.00	33.31
3.52	2.21	-0.56	1.29	1.00	33.30
4.02	2.21	-0.56	1.29	1.00	33.30
4.52	2.21	-0.56	1.29	1.00	33.29
5.02	2.21	-0.56	1.29	1.00	33.29
5.52	2.21	-0.56	1.29	1.00	33.28
6.02	2.21	-0.56	1.29	1.00	33.28
6.52	2.21	-0.56	1.29	1.00	33.27
7.02	2.22	-0.56	1.29	1.00	33.27
7.52	2.22	-0.56	1.29	1.00	33.26
8.02	2.22	-0.56	1.29	1.00	33.26
8.50	2.22	-0.56	1.29	1.00	33.25
9.00	2.22	-0.56	1.29	1.00	33.24
9.50	2.22	-0.56	1.29	1.00	33.24
10.00	2.22	-0.56	1.29	1.00	33.23
10.50	2.22	-0.56	1.29	1.00	33.23
11.00	2.22	-0.56	1.28	1.00	33.22
11.50	2.22	-0.56	1.28	1.00	33.22
12.00	2.22	-0.56	1.28	1.00	33.21
12.50	2.22	-0.56	1.28	1.00	33.21
13.00	2.22	-0.56	1.28	1.00	33.20
13.50	2.22	-0.56	1.28	1.00	33.20
14.00	2.22	-0.56	1.28	1.00	33.19
14.50	2.22	-0.56	1.28	1.00	33.19
15.00	2.22	-0.56	1.28	1.00	33.18

Figure D11 Sample Output File (continued)

References

1. Moncarz, H.T., Bernard, J.E., Fancher, P.S., A Simplified, Interactive Simulation for Predicting the Braking and Steering Response of Commercial Vehicles, Highway Safety Research Institute/University of Michigan, Ann Arbour, Michigan, August 1975, UM-HSRI-PF-75-8.

2. Bernard, J.E., Kim, T.M., Modified Articulated Vehicle Simulation, Highway Safety Research Institute/University of Michigan, Michigan, April 1977, UM-HSRI-76-23.

3. Snelgrove, F.B., Billing, A.M., Choi, C., Performance Evaluation of Several Jackknife Control Devices, Ministry of Transportation and Communications, Ontario, August 1980, CVOS-TR-80-03.

4. Billing, A.M., Visentin, A., Validation of a Computer Model of Tractor-Semitrailer Handling That Includes Five Jackknife Control Devices, Ontario Ministry of Transportation and Communications, April 1978, CVOS-TR-78-07.

5. El-Gindy, M., Wong, J.Y., A Comparison of Various Computer Simulation Models for Predicting the Directional Responses of Articulated Vehicles, Swets & Zeitlinger, Vehicle Systems Dynamics, 1987, pp 249-268.

6. Faucher, P.S. Ervin, R.D., Winkler, C.B., Gillespie, T.D., Factbook of the Mechanical Properties of the Components for Single-Unit and Articulated Heavy Trucks, University of Michigan Transportation Research Institute, Ann Arbour, Michigan, December 1986, UMTRI-86-12, DOT-HS-807-125.

APPENDIX "E"
Steady State Handling Characteristics

APPENDIX "E" STEADY STATE HANDLING CHARACTERISTICS

A vehicle travelling at a constant forward speed in a steady flat turn, is a basic (and simple) example of the movement of a vehicle that is suitable for examination of handling characteristics (1). Investigation of the steady state handling characteristics of the vehicle was very important in a study such as this. The information gained aided in predicting and understanding the outcome of the configurations undergoing the specified manoeuvres. Development of articulation angle response curves and handling diagrams provided insight into the handling tendencies of the configurations and their characteristic instabilities. The handling diagrams were also required to determine the steer angle input to the program.

For ease of understanding, a simplified model of the tractor semitrailer was analyzed. This simplified model, referred to as a "bicycle model", is depicted in Figure E1. This approach to analysis of the vehicle characteristics was previously used by a number of authors, including Passerello (2) and El-Gindy (3). The assumptions associated with the model were:

1) Motion was limited to the horizontal plane,
2) The radius of the turn was assumed very large with respect to the tractor wheelbase, L,
3) The steer angle, δ was small
4) The slip angle, α was small
5) The forward velocity was constant
6) The aligning torque was constant, and
7) The tandem suspensions were approximated by nearly equivalent single-axis suspensions.

From geometry, the relationship between the front wheel slip angle, α_{11}, the rear wheel slip angle, α_{12}, the steer angle, δ, the radius of the turn, R, and the tractor wheel base length, L, for this model is:

$$\alpha_{11} = \alpha_{12} + \delta - \frac{L}{R} \qquad [1]$$

The conditions of lateral and vertical equilibrium required that the following relationships be satisfied.

$$\frac{Fy_{11}}{Fz_{11}} = \frac{Fy_{12}}{Fz_{12}} = \frac{\bar{V}^2}{Rg} \qquad [2]$$

A plot of $\bar{V}^2/Rg$ versus $\alpha_{12} - \alpha_{11}$, Figure E2, was constructed from equation [14] and the normalized side force versus slip angle curve for a tire pair (as shown in Figure E3).

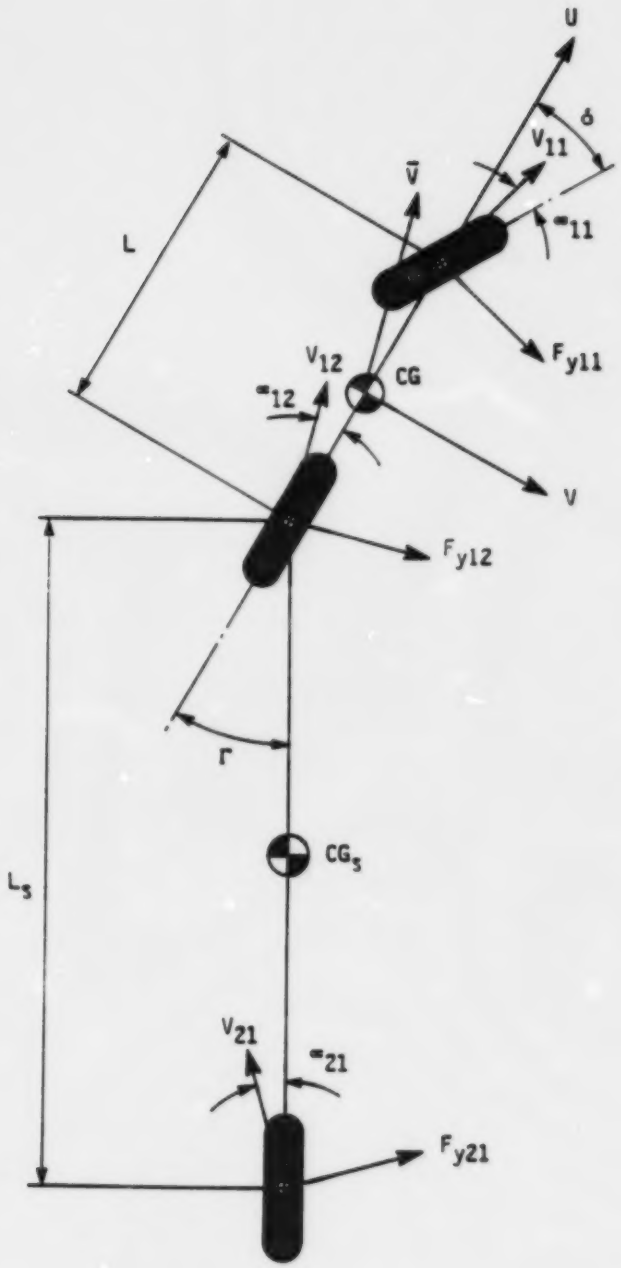


Figure E1 Simplified Tractor Semitrailer Model

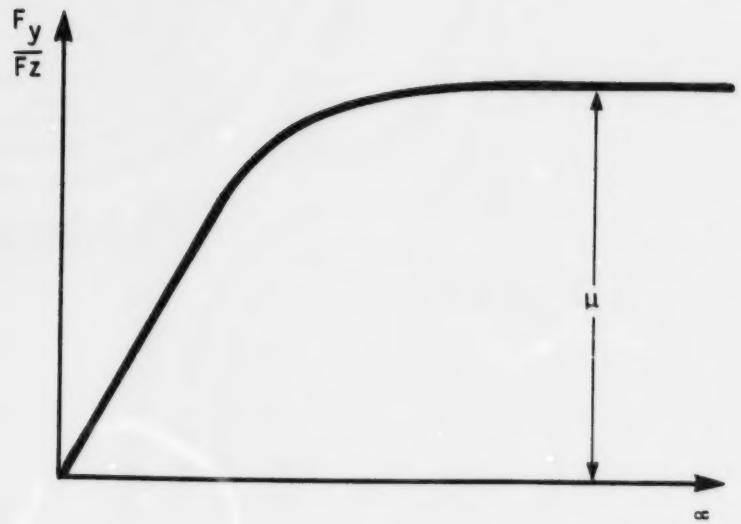


Figure E2 Construction of the Handling Curve

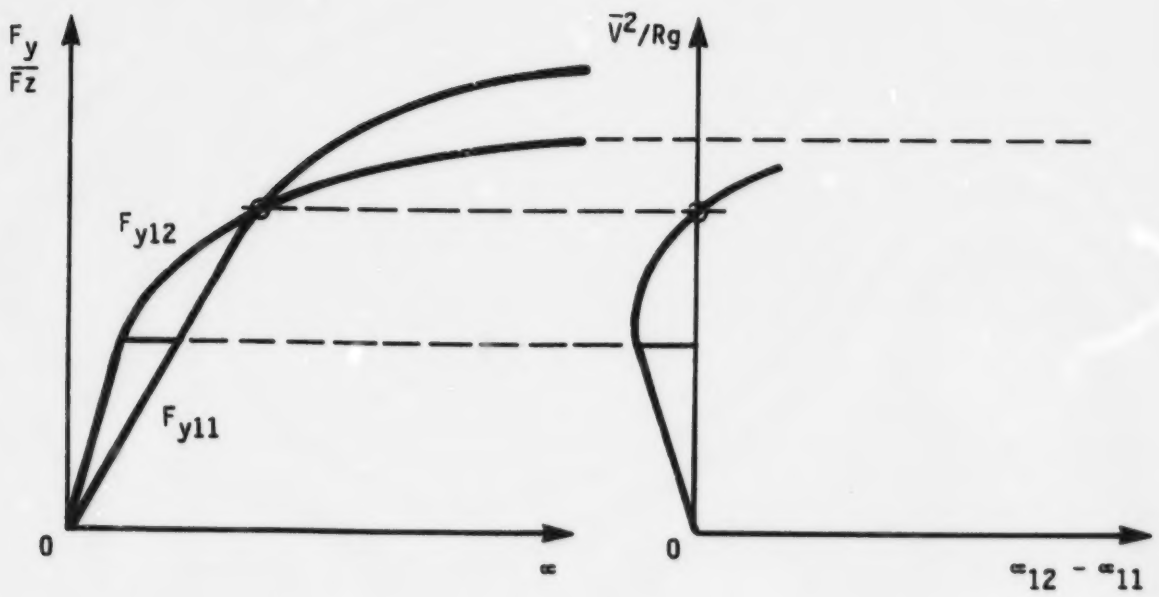


Figure E3 Normalized Side Force Characteristics

Equation [1] was rewritten in the form;

$$\alpha_{12} - \alpha_{11} = \frac{L}{R} - \delta \qquad [3]$$

Knowing;

$$R = \frac{U}{r} \qquad [4]$$

Substitutions into equation [3] provided;

$$\alpha_{12} - \alpha_{11} = \frac{rL}{U} - \delta \qquad [5]$$

When $\bar{V}^2/Rg$ was plotted against L/R for different values of U, a family of curves showing lines of constant velocity was obtained as shown in Figure E4. Combination of Figure E2 and Figure E4, (with equation [5] substitution) resulted in the "Handling Diagram" of the configuration (Figure E5). This diagram graphically enabled the determination of the required steer angle, δ, of the system executing a turn of radius R at speed U. For example, for a known radius the abscissa (L/R) at point 'A' was entered. A vertical line from this point was followed upward until it intersected the constant velocity line corresponding to the intended speed, point B. The horizontal distance from this point to the handling curve, point C, (constant lateral acceleration, V^2/Rg) provided the required steer angle, δ, for the manoeuvre.

Other information extracted from the handling diagram was the handling tendencies of the configuration (i.e. was the vehicle oversteer, neutralsteer, or understeer). A vehicle was considered to be understeer when the required steer angle, δ, to negotiate a steady turn of radius R increased with an increased forward velocity, U. Conversely, a vehicle was considered oversteer when the steer angle, δ, decreased with increasing forward velocity U. Neutralsteer occurred when δ was independent of U.

The slope of the handling curve indicated these handling tendencies of the configuration. A negative slope therefore indicated understeer, a positive slope corresponded to oversteer, and an infinite slope indicated neutralsteer. Referring to Figure E5, point 'D' separated the understeer region from the oversteer region. A vehicle with this type of handling diagram, negotiating a constant radius turn, required an increasing steer angle, δ, to point 'D' and a gradually decreasing steer angle beyond point 'D' as the vehicle speed increased.

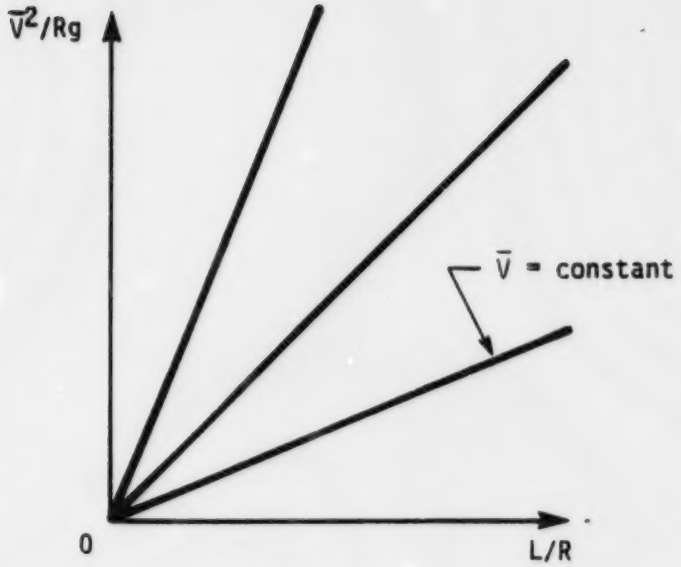


Figure E4 Lines of Constant Velocity

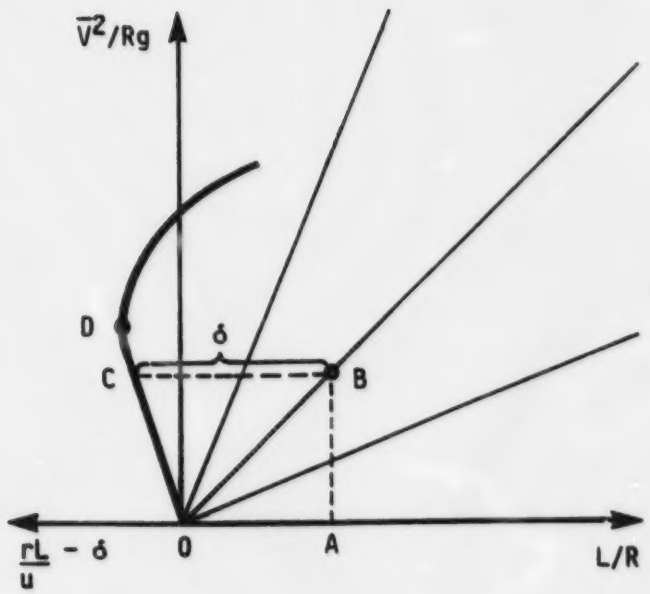


Figure E5 The Handling Diagram

A simple method for determining the divergent instability of the tractor yaw rate, Ψ, (jackknifing instability) with the aid of the handling diagram was set forth by Pacejka and quoted by Passerello (2). The method was based on the fact that for stability:

$$\frac{\delta\ (\delta)}{\delta\ (L/R)} > 0 \qquad [6]$$

Hence the critical lateral acceleration on a line of constant velocity beyond which instability occurred can be found by determining the point on the handling diagram where its tangent was parallel to the constant velocity line being considered. Connecting these critical points on all the constant velocity lines yielded the "stability boundary" (Figure E6).

Another important aspect of the steady state analysis was the investigation of the articulation angle response characteristics (Γ/δ versus U). As reported by Bourassa (8), R.D. Ervin stated that the steer angle, δ, and the articulation angle, Γ, could be expressed as:

$$\delta = \frac{L}{R} + \Omega \frac{\bar{V}^2}{Rg} \qquad [7]$$

$$\Gamma = \frac{L_s}{R} + \Omega_s \frac{\bar{V}^2}{Rg} \qquad [8]$$

$$\Omega = \frac{F_{z11}}{C_{\alpha 11}} - \frac{F_{z12}}{C_{\alpha 12}} \qquad [9]$$

$$\Omega_s = \frac{F_{z12}}{C_{\alpha 12}} - \frac{F_{z21}}{C_{\alpha 21}} \qquad [10]$$

where Ω and Ω_s corresponded to the understeer or oversteer gradients for the tractor and semitrailer respectively. A negative value indicated an oversteer vehicle, while a positive value corresponded to understeer.

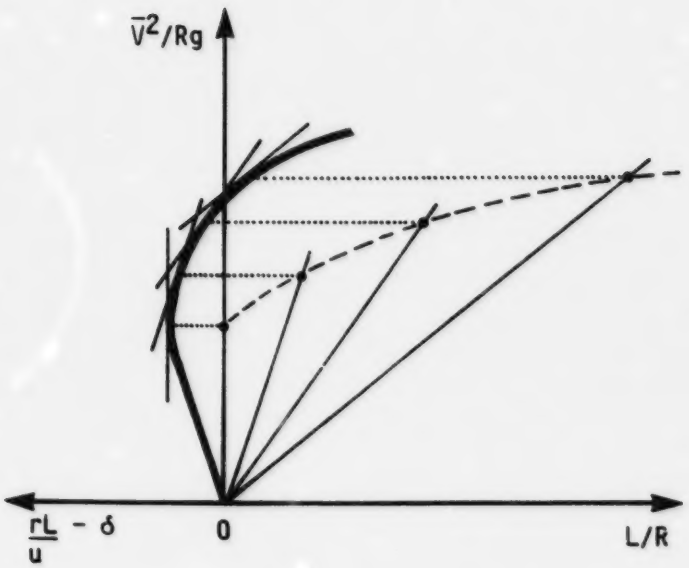


Figure E6 Illustration of the Vehicle Stability Boundary

Dividing equation [8] by equation [7] yielded the following relation for articulation angle gain:

$$\frac{\Gamma}{\delta} = \frac{L_s + \Omega_s \bar{V}^2/g}{L + \Omega \bar{V}^2/g} \qquad [11]$$

The values of L, L_s, Ω and Ω_s provided the stability characteristics of a particular configuration. Some general observations drawn from equation [11] were:

- at low speeds the articulation angle gain, Γ/δ, was approximately equal to L_s/L
- at high speeds Γ/δ approached Ω_s/Ω
- Γ/δ passed through zero going from positive to negative if $U_{cs} = (-L_s g/\Omega_s)^{\frac{1}{2}}$ $(\Omega_s < 0)$
- Γ/δ was nearly infinite if U was approximately equal to $U_c = (-Lg/\Omega)^{\frac{1}{2}}$ $(\Omega < 0)$

Speeds greater than the critical speed, U_c, resulted in vehicle instability when the understeer gradient for the tractor was less than zero $(\Omega < 0)$, i.e. the tractor was oversteer.

Six possible modes of tractor semitrailer articulation angle response were achieved based on the understeer gradient ratios Ω_s/Ω and the wheel base ratios L_s/L (Figure E7). Cases E7(A), E7(B) and E7(C) illustrate statically stable conditions while E7(D), E7(E) and E7(F) indicate various forms of instability. Cases E7(D) and E7(E) show jackknife tendencies as indicated by a positively increasing gain as U approached U_c. In case E7(F) the gain goes through zero and approaches negative infinity as U approaches U_c, indicating that the trailer swings further out of lane as U approaches U_c, thus displaying a trailer swing type of instability.

Characteristics of Each Configuration

Approximately 130 simulation runs were required to determine the steady-state handling characteristics of the four configurations, both laden and unladen. The steady-state behaviour is an important aspect of vehicle handling studies. Its role in this study was twofold. First, the general handling tendencies of the vehicles were displayed. This indicated any inherent instabilities of the combinations, and gave an idea as to the relative performance of the vehicles during particular manoeuvres. Second, construction of handling diagrams enabled the determination of the steer angle required to complete a desired steady-state manoeuvre, such as the constant radius turn.

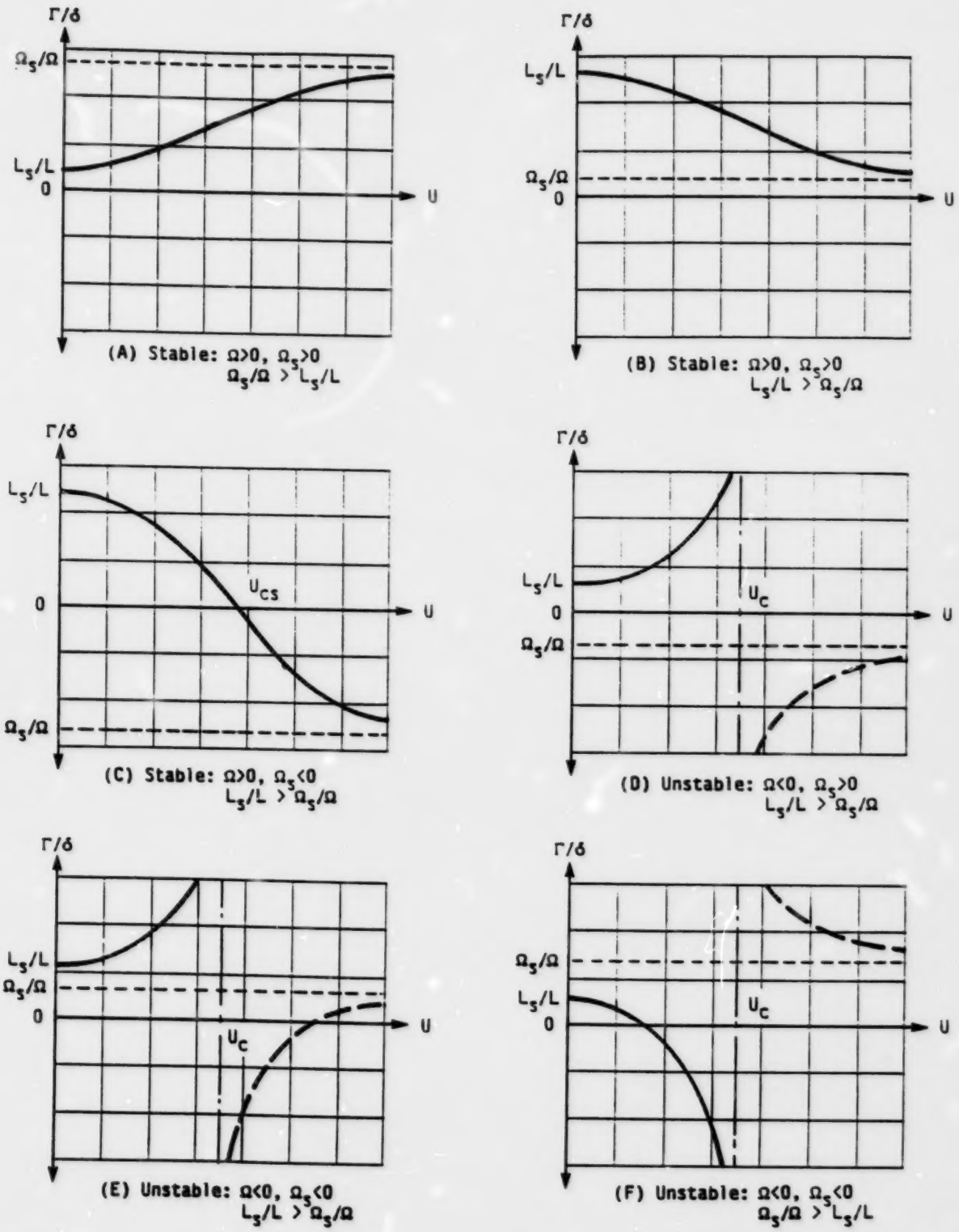


Figure E7 Cases of Tractor Semitrailer Handling Behaviour

Figure E8 showed the articulation response curves of the four combinations (refer to Section 5.3.1 for vehicle parameters), in both their empty and loaded states. Comparison of these curves with the characteristic curves of Figure E7 showed the relative stability, or instability of the vehicles.

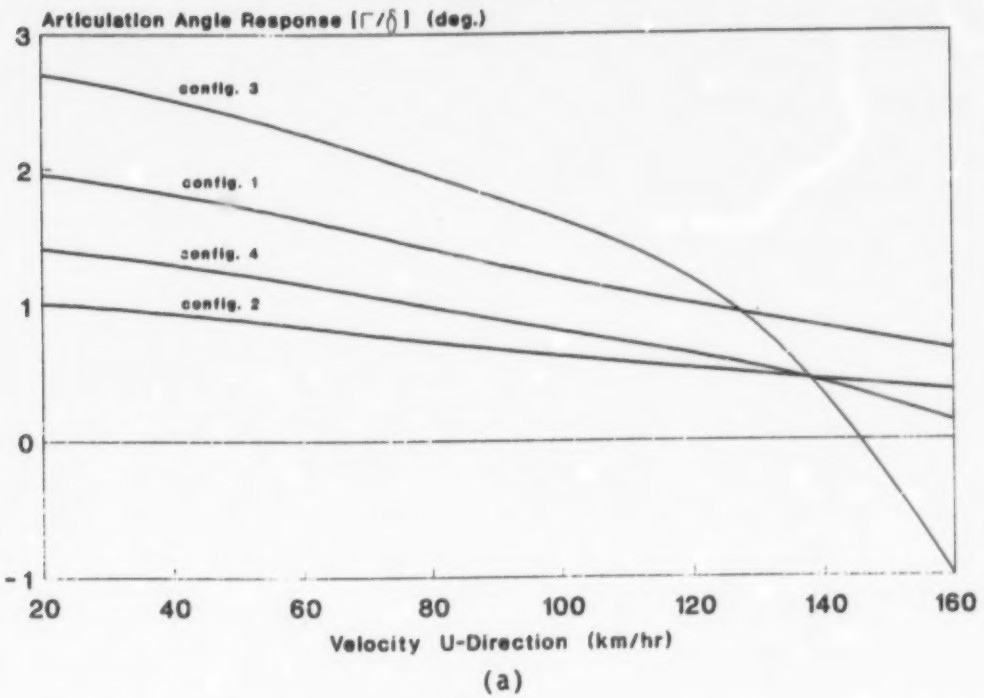
The most obvious observation from Figure E8(a) was the instability of configuration 3 as it approached its critical velocity, U_c, of approximately 160 km/hr. This form of instability, as presented in Figure E7(F), was characterized by the vehicles tendency to undergo trailer swing as U approached U_c. Configuration 4 loaded behaves in a similar but less extreme manner to configuration 3 (recall configurations 3 and 4 were both comprised of the same 3.6 m wheel base tractor). Configurations 1 and 2 displayed stability in the form of Figure E7 (B). The empty configurations, Figure E8(b), exhibited characteristics that were slightly more stable than the loaded configurations. Again, Figure E7(B), displayed the type of stability characteristic which are typical of all empty configurations.

The ordinate on the articulation angle response curves closely corresponded to the theoretical values of L_s/L, as shown in Figure E7. The ratio of the non-linear understeer gradients of the tractor and semitrailer, Ω_s/Ω, were determined by the corresponding asymptote of the articulation angle gain in Figure E8. It should be noted that the values for Ω_s and Ω, and thus the ratio Ω_s/Ω, evident in Figure E8, incorporated the non-linear tire characteristics from the computer code.

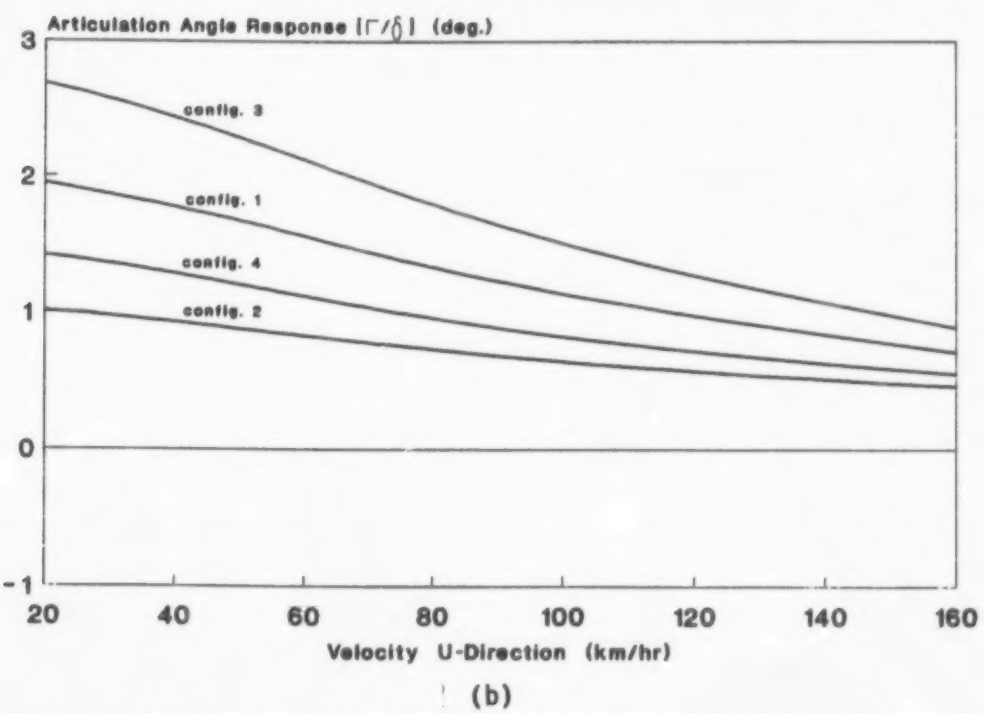
The handling diagrams (Figure E9 (a) through (h)) allowed the determination of the required steer angle to input into the program. Also evident from the handling diagrams were some of the handling characteristics of the vehicles. For example, all four configurations, when loaded, exhibited a transition from understeer to oversteer at lateral accelerations of approximately 2.9 m/s^2 (0.3 g). Conversely the empty configurations remained continuously understeer for lateral accelerations as high as 4.5 m/s^2 (0.46 g).

The tractor played a dominant role in the form of the handling diagram since $rL/U - \delta$ and L/R was mainly tractor characteristics. This was evident by comparing Figure E9(a) and (b) or (e) and (f) with (c) and (d) or (g) and (h) respectively. The longer wheelbase tractor, configurations 1 and 2, were more understeer and slightly more stable than the shorter wheelbase tractor, configurations 3 and 4. Similarly, the degree of oversteer above the 0.3 g point was greater for the shorter wheelbase tractor, configurations 3 and 4, than the longer wheelbase tractor, configuration 1 and 2.

The impact of the trailer on the handling characteristics was made evident by comparing configuration 1 with configuration 2 and configuration 3 with configuration 4. The shorter trailer, configurations 2 and 4, showed slightly more stability when joined to either tractor than the longer trailer, configurations 1 and 3. The effect of payload was also evident when the respective handling diagrams of the empty and loaded configurations in Figure E9 were compared. From these diagrams it became apparent that an increase in trailer payload tended to increase the instability of a vehicle.

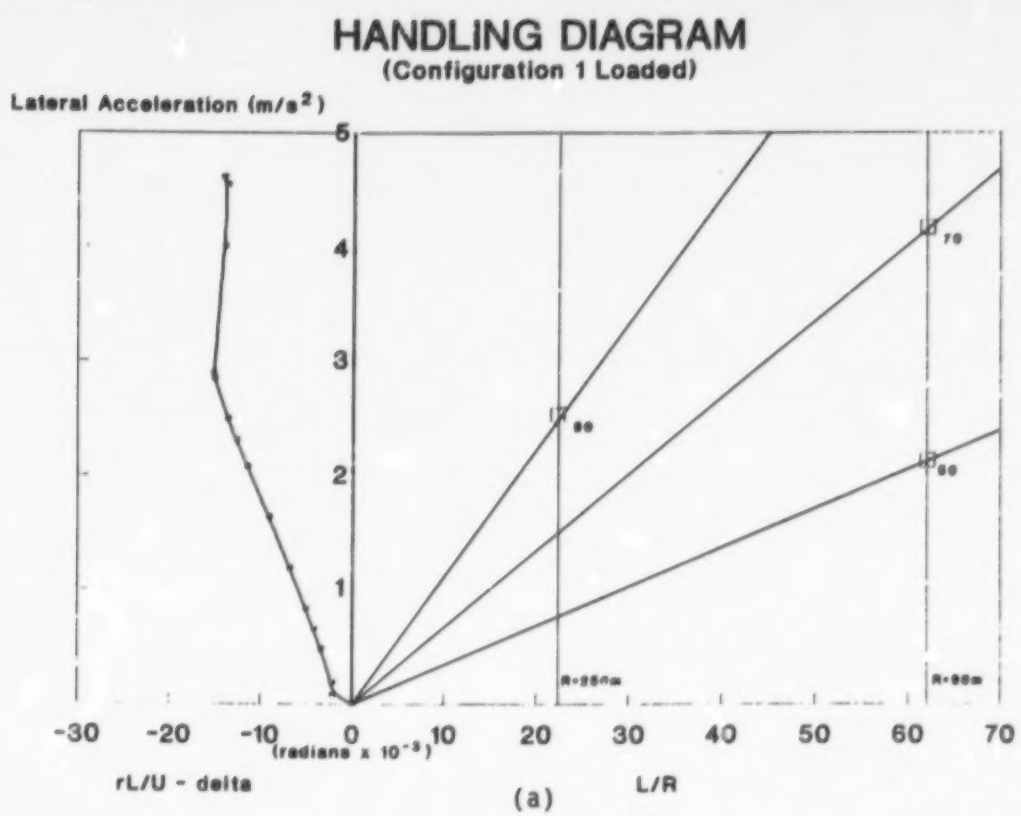


(a)



(b)

Figure E8 Articulation Angle Response Curves of Simulated Vehicles



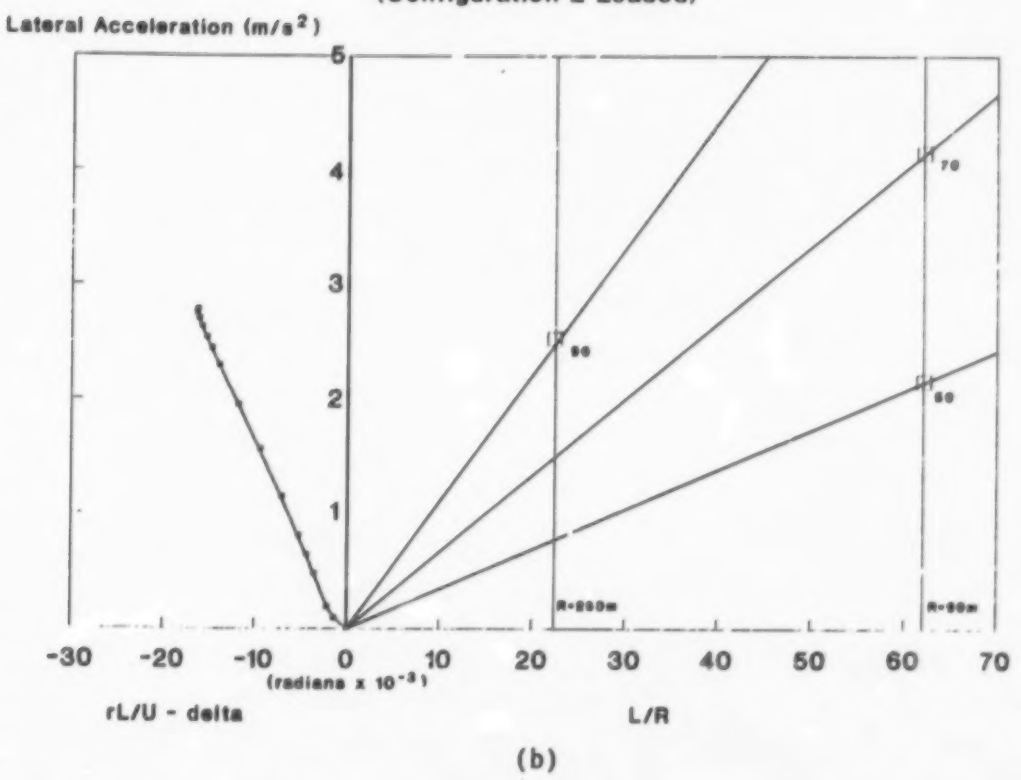
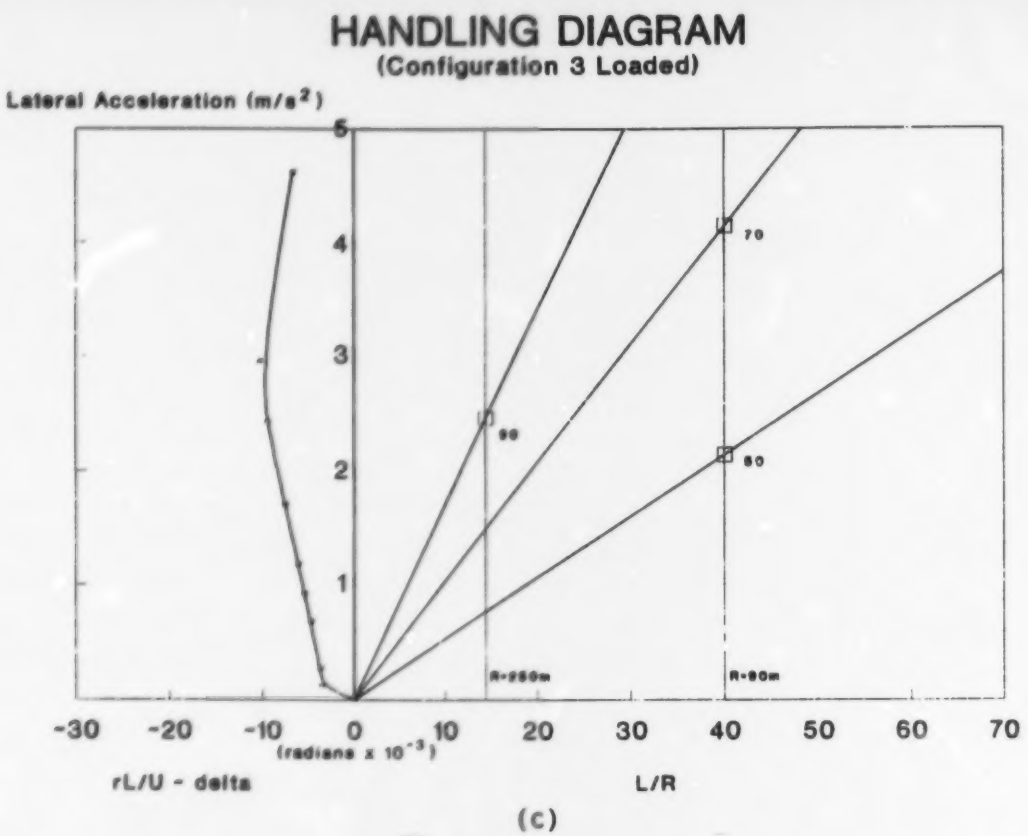
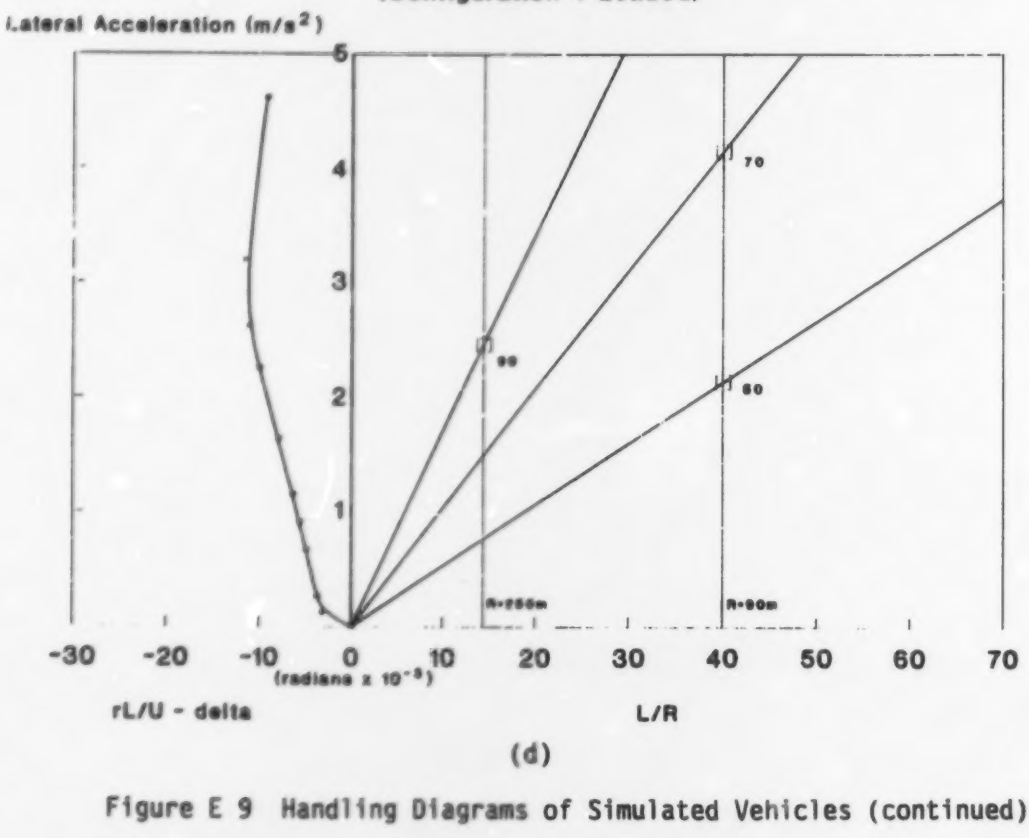


Figure E9 Handling Diagrams of Simulated Vehicles

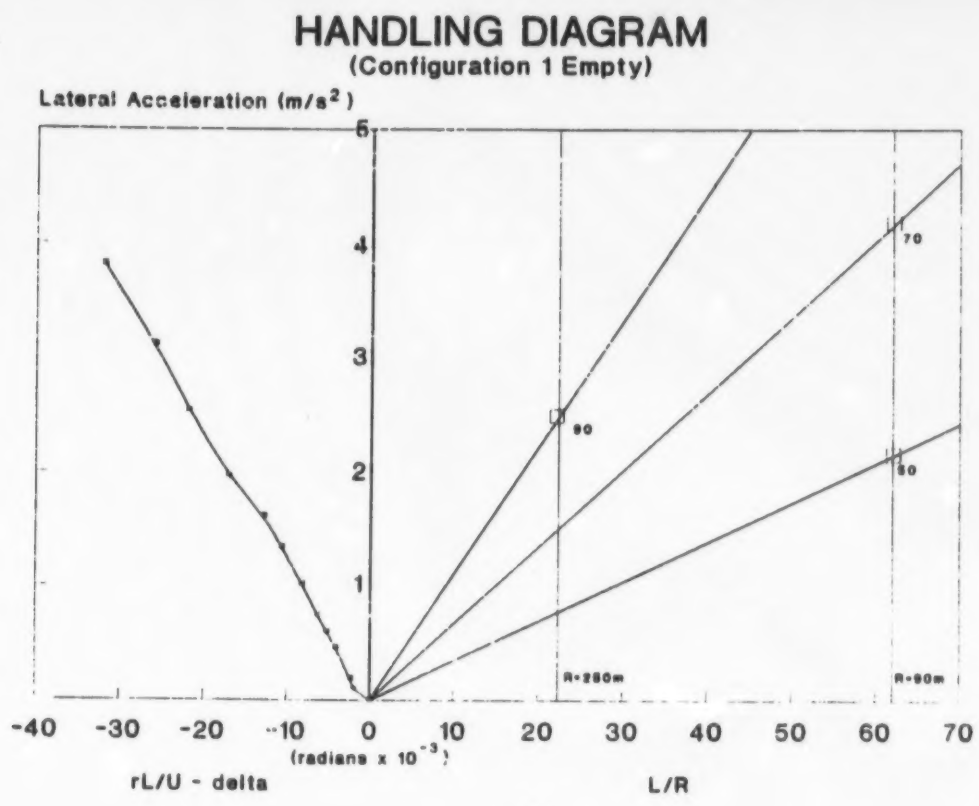


(c)

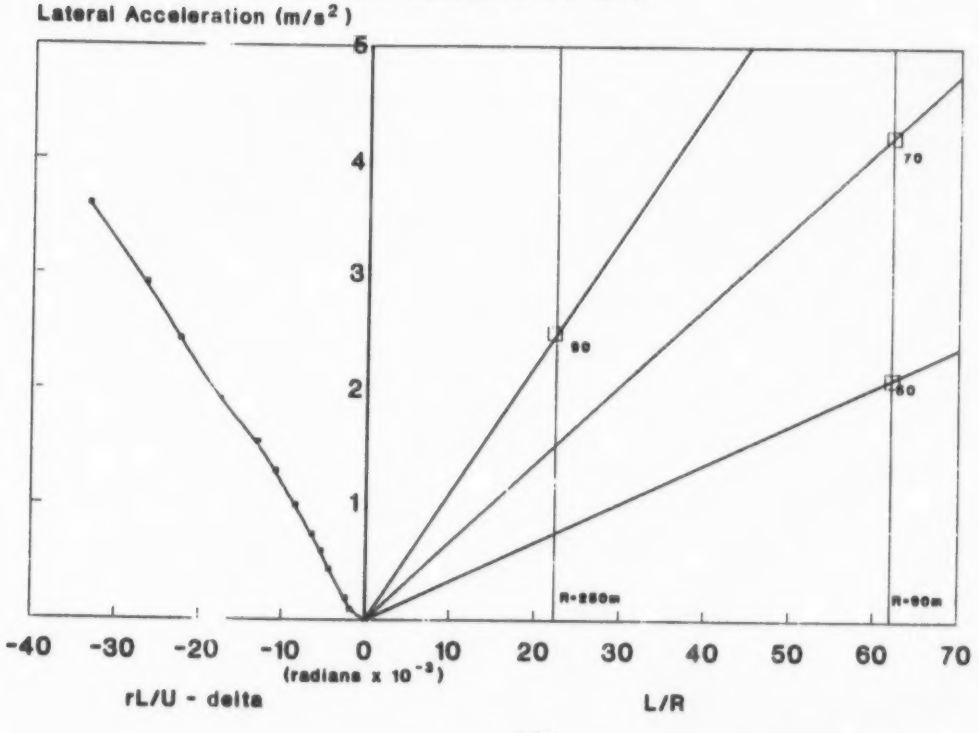


(d)

Figure E 9 Handling Diagrams of Simulated Vehicles (continued)

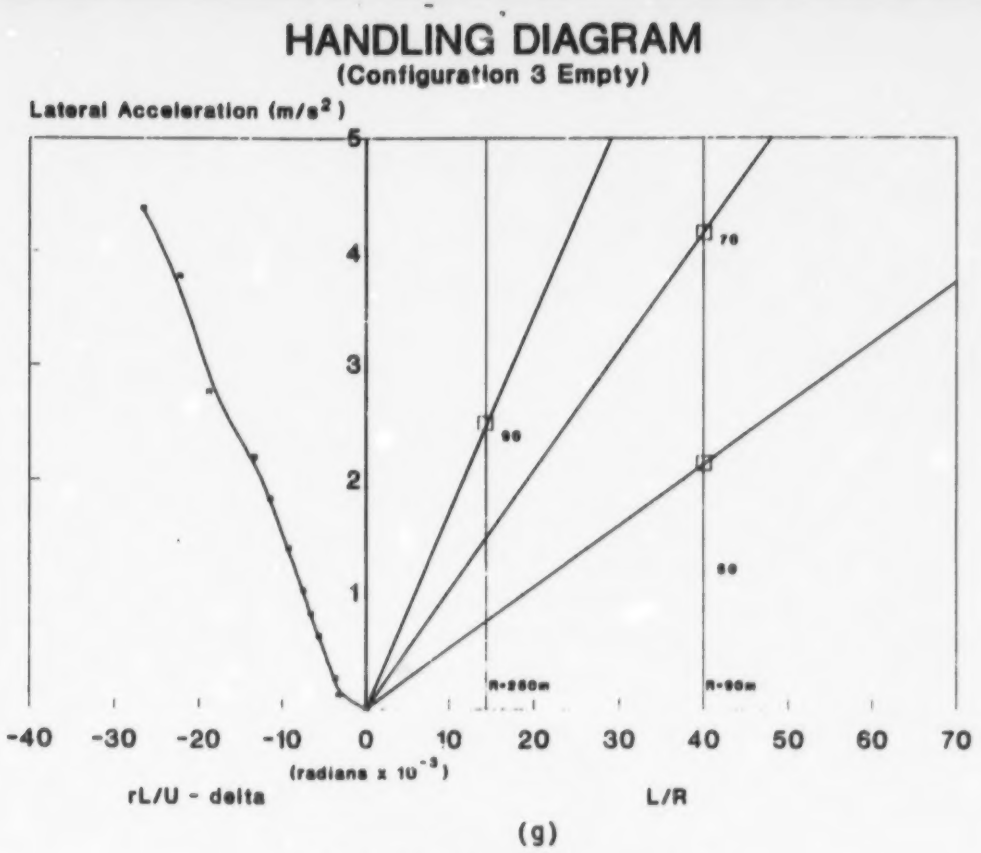


(e)

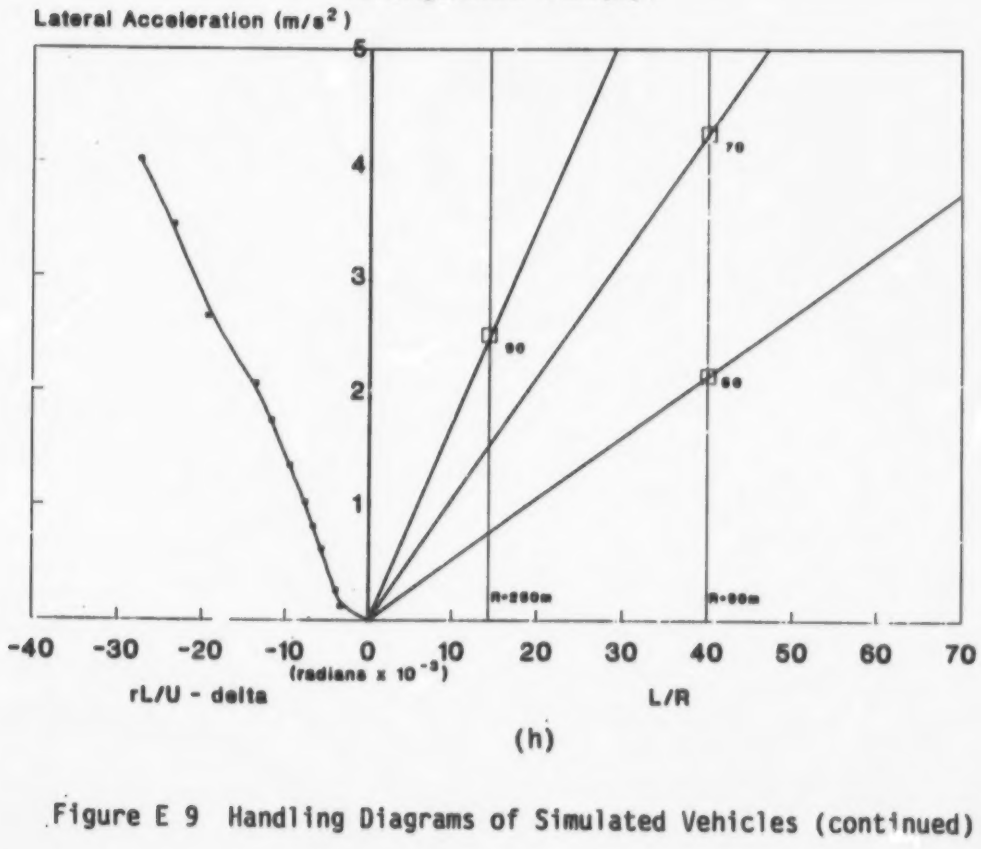


(f)

Figure E 9 Handling Diagrams of Simulated Vehicles (continued)



(g)



(h)

Figure E 9 Handling Diagrams of Simulated Vehicles (continued)

References

1. Vik, F., Handling Performance of Truck-Trailer Vehicles: A State-of-the-Art Survey, International Journal of Vehicle Design, Vol. 6, No. 3, U.K., 1985.

2. Passerello, C.E., A Simplified Analysis of the Steady-State Turning of Articulated Vehicles, Society of Automotive Engineers, 1979.

3. El-Gindy, M., Wong, J.Y., Steering Response of Articulated Vehicles in Steady-State Turns, Society of Automotive Engineers, Chicago, December 1985, SAE852335.

APPENDIX "F"

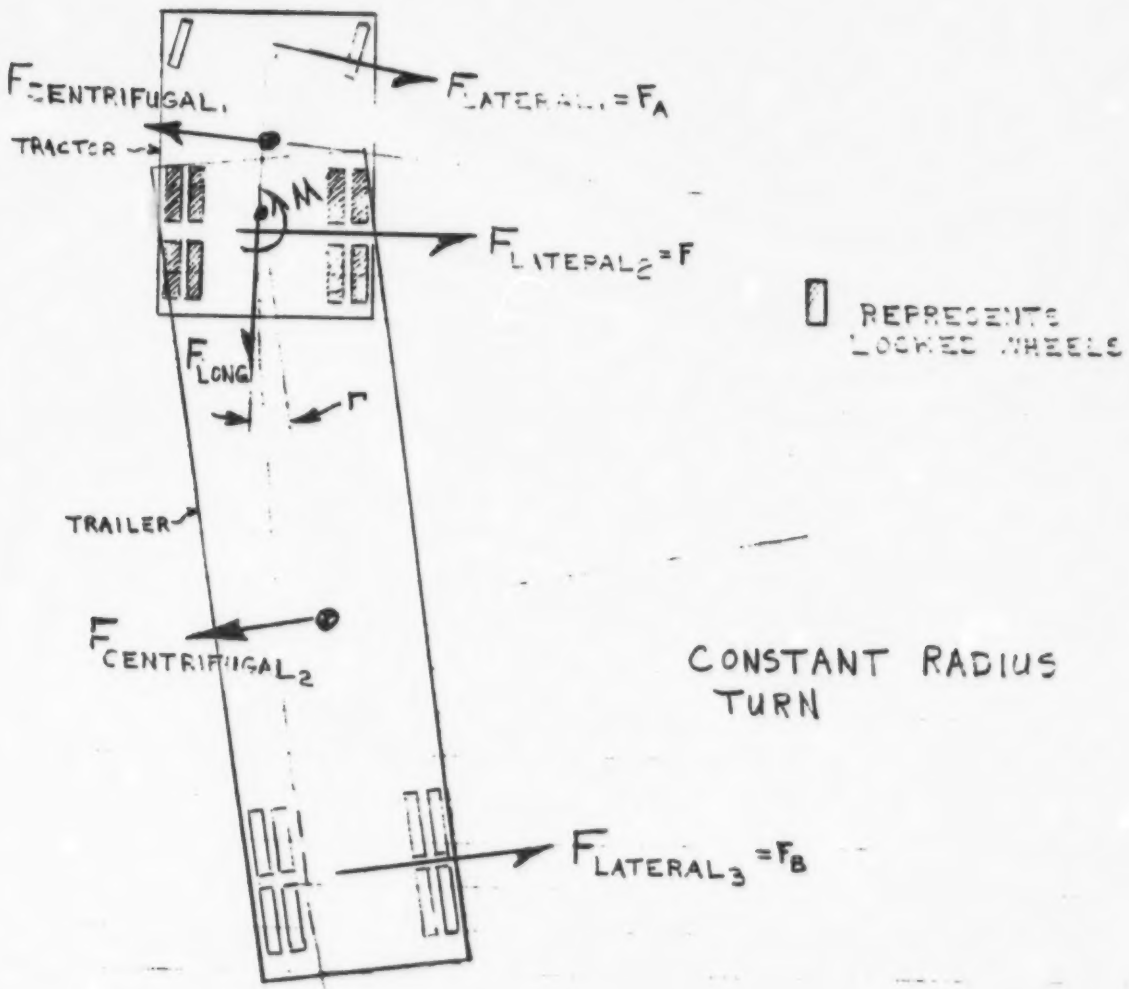
"Theoretical Torques Produced in Jackknife and Trailer Swing"

Anti-Jackkifing Analysis - Examination of Tractor Jackknife

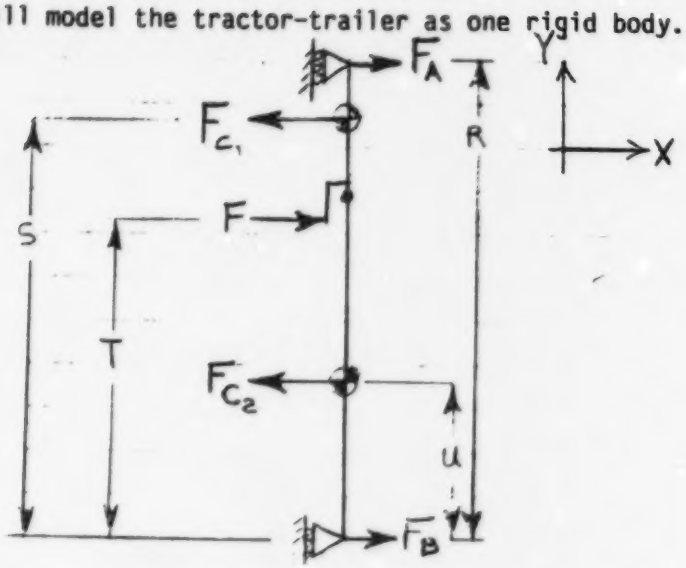
Purpose: To determine the approximate level of resistive torque an AJS would need to provide to prevent jackknifing.

Solution: Begin by constructing a F.B.D. of the tractor and trailer, where the two are locked together at the 5th wheel to simulate an AJS and the tractor rear wheels are locked due to braking.

F.B.D



I will model the tractor-trailer as one rigid body.



Assume the articulation angle is small and assume tractor-trailer is one rigid body.

For Static Equilibrium in the X-Direction

$\Sigma F_x = 0$

$\therefore F_a + F_b = F_{c1} + F_{c2} - F$

Solving for $F_a + F_b$

Tractor #1:

Where V = 30 mph (45.5 ft/sec)

$M = W/g = \frac{19,945 \text{ lbs}}{32.2 \text{ ft/sec}^2}$

r = 295 ft.

$F_{c1} = M_{tractor} \cdot \frac{V^2}{r} = \frac{19,945 \text{ lbs}}{32.2 \text{ ft/sec}^2} \cdot \frac{(45.5)^2}{295} = 4,347 \text{ lbs}$

F≈ 67 lbs (From Simulation Data)

Trailer #1: (empty)

$F_{c1} = M_{Trailer} \cdot \frac{V^2}{r} = \frac{14,760 \text{ lbs}}{32.2 \text{ ft/sec}^2} \cdot \frac{(45.5)^2}{295} = 3,217 \text{ lbs}$

$\therefore F_a + F_b = F_{c1} + F_{c2} - F$

$= 4,347 + 3,217 - 67$

$\therefore F_a + F_b = 7,288 \text{ lbsf} \qquad (1)$

Examining Fifth Wheel Torque required, Using a loaded Trailer

Trailer #1

m_t = 19,867 kg
= 43,799 lbsm
= 1,360 slugs

where $F_c = M_t \cdot \frac{V^2}{r} = 1{,}360 \cdot \frac{(45.5)^2}{295} = 9{,}544$ lbsf

From before

$F_a + F_b = F_{c1} + F_{c2} - F$
$= 4{,}347 + 9{,}544 - 67$ (Tractor #1)
$\therefore F_a + F_b = 13{,}824$ lbsf (2)

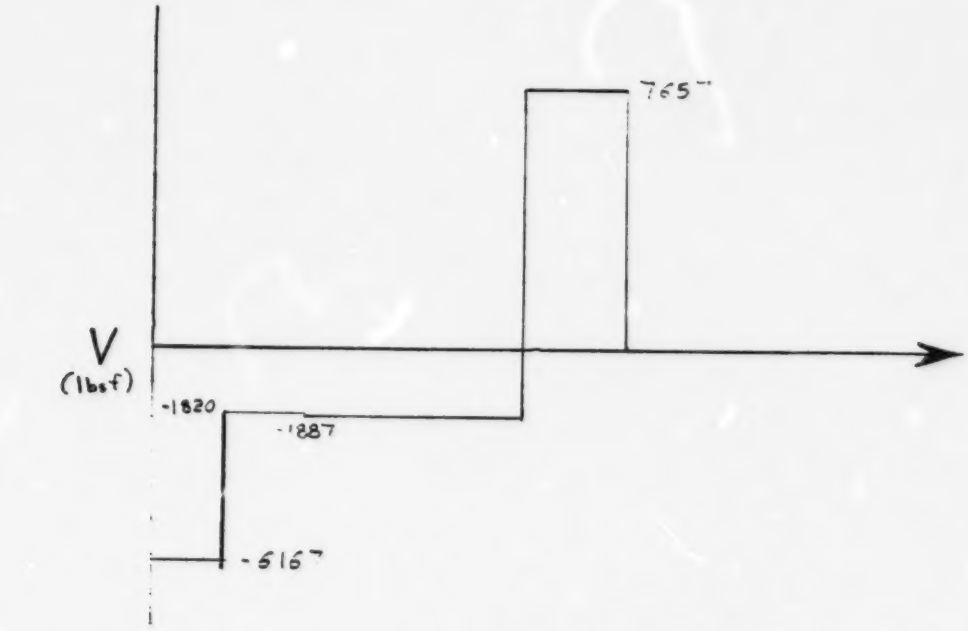
Looking at Moments

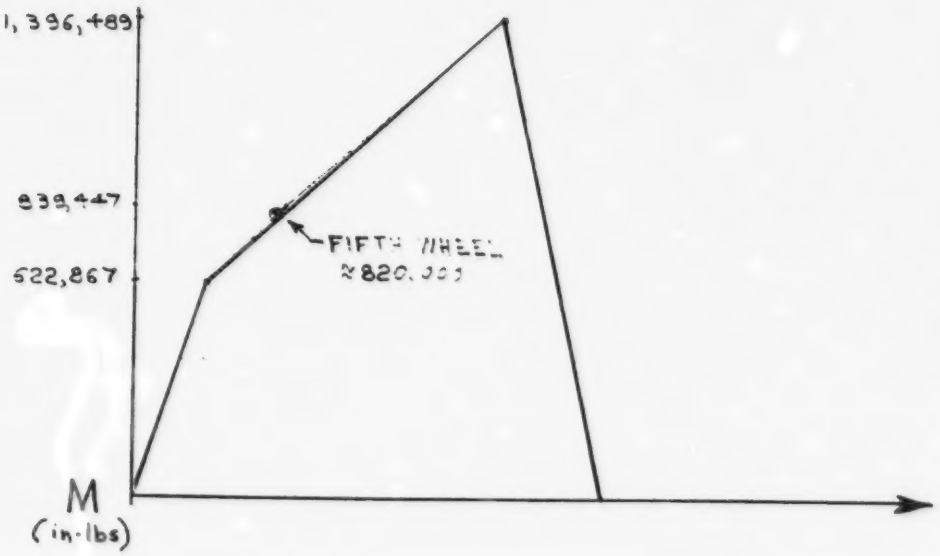
$\Sigma M = 0$ (for Static Equilibrium)
$F_a \cdot R + F \cdot T - F_{c1} \cdot S - F_{c2} U = 0$

$F_a \cdot 697.6 + 67 \cdot 477.6 - 4347 \cdot 596.6 - 9{,}544 \cdot 182.4 = 0$

$\therefore F_a = 6{,}167$ lbsf
Substituting into Eqn (2) $\Rightarrow F_b = 7{,}657$ lbsf

Creating Shear Diagrams & Moment Diagrams





From the Moment Diagram the torque required at the Fifth Wheel to maintain a rigid body (ie. No Articulation) is approximately 820,000 in·lbs (= 92,600 N·m) for the loaded (13,600 kg) trailer case.

Repeating the same process for an empty trailer resulted in a resistive torque of 460,000 in·lb (52,000 N·m)

Calculation of Moment Required to Stop Articulation for various stages into Jackknife (ie. $\Gamma = 5°, 10°, 20°, 30°$).

This is extremely difficult to do, due to dynamic conditions, but will approximate as follows:

$$M_{AJS} \approx \text{Torque}_{\text{At } 0° \text{ Jackknife}} - I_{\text{Tractor}} \cdot \frac{(\dot{\Gamma}_f - \dot{\Gamma}_i)}{\Delta t} + \text{Torque due to friction at front wheels.}$$

Where $\text{Torque}_{\text{At } 0° \text{ Jackknife}}$ = Previously calculated torque (ie. Moment due to centrifugal force).

and

$I_{\text{Tractor}} \cdot \frac{(\dot{\Gamma}_f - \dot{\Gamma}_i)}{}$ represents the torque required to reduce the tractor inertia to zero. Where $\dot{\Gamma}_f = 0$ deg/sec

$\dot{\Gamma}_i = \dot{\Gamma}_{\Gamma/2}$

Δt = time from $\dot{\Gamma}_{\Gamma/2}$ to $\dot{\Gamma}_f$

and

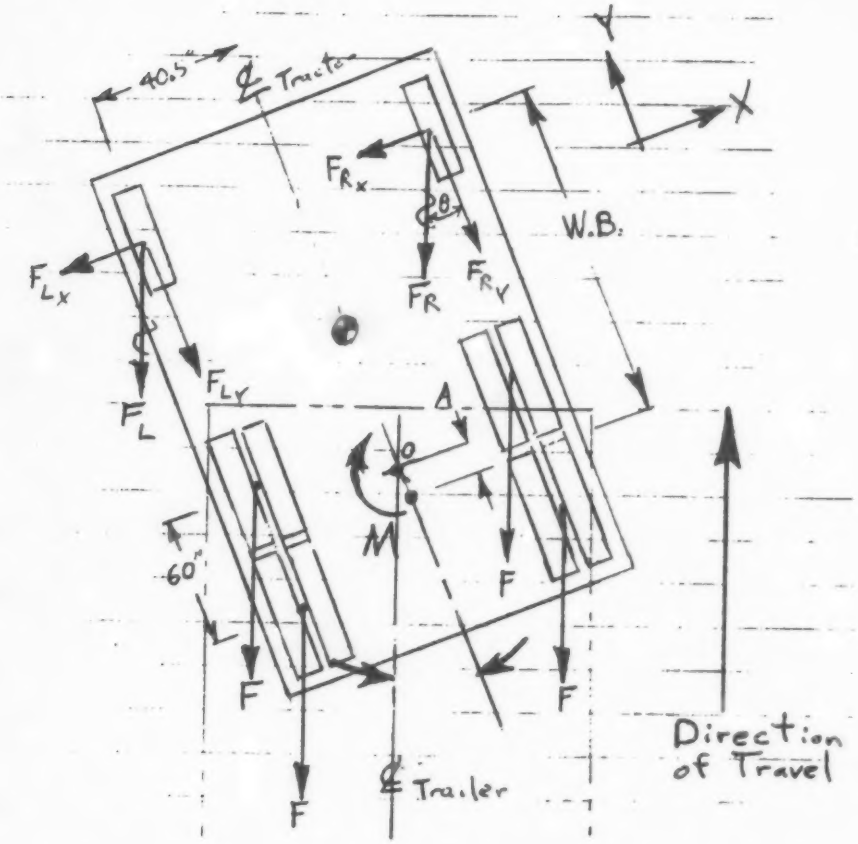
Torque due to friction at front wheels.

$= F_L \cdot \sin\emptyset \cdot (W.B. - A) \cdot 2$
$- F \cdot \sin\emptyset \cdot (30 + A) \cdot 2$
$+ F \cdot \sin\emptyset \cdot (30 - A) \cdot 2$
(See F.B.D next page for derivation).

Where $F_L = N_{Front}\, u_{Front}$
and $F = N_{Rear} \cdot u_{Rear}$

Let $u = 0.25 = u_{Rear} = u_{Front}$
$W.B. = 220$ in.
$N_{Front} = 10{,}790$ lbs
$N_{Rear} = 9{,}155$ lbs

F.B.D.of Tractor



Where W.B. = Tractor Wheelbase

Γ = Articulation Angle

A = Distance from L of Rear Axles to fifth wheel

O = Fifth wheel pivot point

M = AJS resistive Torque

Assumptions: Front wheels do not rotate (ie. locked).
Front wheels remain in a straight ahead position.
All wheels at the rear have the same normal load and friction coefficient as do the front wheels.

F = Friction Force at each set of rear wheels

$F_L = F_R$ = Friction Force at front wheels

Calculating Tractor Inertia about Fifth Wheel (Empty Trailer Case).

From Parallel Axis Theorem $I_{5th\ Wheel} = I_{C.G.} + md^2$

$= 62{,}000 + 9{,}047 \cdot (2.56)^2$

$= 121{,}000\ kg \cdot m^2$

Calculating

$$M_{AJS_{5°}} \approx 460{,}000\ in \cdot lbs - 121{,}000 \frac{(0-0.14\ rad/sec)}{0.63\ sec} \cdot \frac{(39.37)}{4.448} + 58{,}905\ in \cdot lbs$$

Values Derived From simulation Data — Calculated from Equation

$\approx 757{,}000\ in \cdot lbs$

$\approx 85{,}000\ N \cdot m$

Repeating this process for larger articulation angles yielded the following.

M_{AJS} @ 10° $\approx 118{,}000\ N \cdot m$

M_{AJS} @ 20° $\approx 96{,}000\ N \cdot m$

M_{AJS} @ 30° $\approx 94{,}000\ N \cdot m$

As the method used to derive these values is quite crude, these values are approximate only.

TRAILER SWING CALCULATIONS

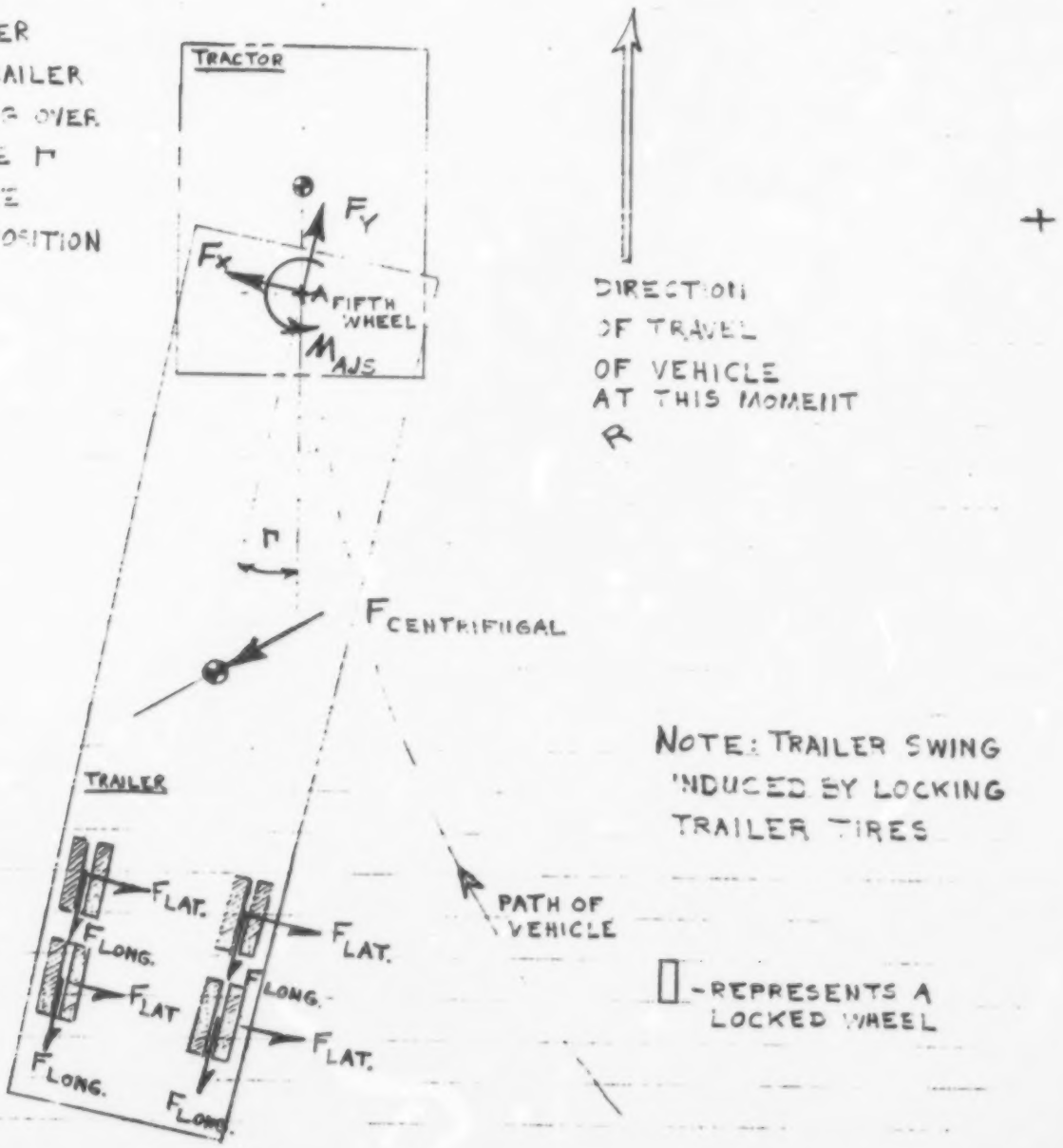
Anti-jackknifing analysis - Examination of Trailer Swing

Purpose: To determine the approximate level of resistive torque an AJS would need to provide to prevent trailer swing.

Solution: Begin by constructing a F.B.D. of the trailer.

F.B.D.

OF TRAILER AFTER TRAILER HAS SWUNG OVER AN ANGLE Γ FROM THE INLINE POSITION

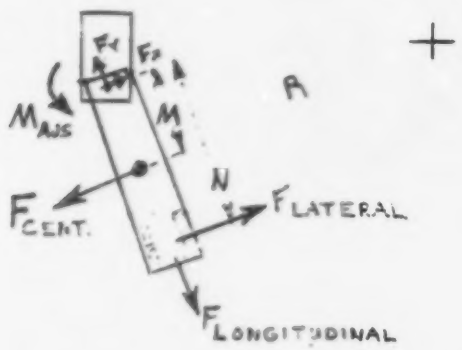


NOTE: TRAILER SWING INDUCED BY LOCKING TRAILER TIRES

▯ - REPRESENTS A LOCKED WHEEL

Examining the Resistive Moments required by the AJS

- To prevent the trailer encroaching into the path of oncoming traffic no trailer swing is desirable.
- Begin by examining the torque required at a swing angle = 0° (ie. no swing)



Solution:

If no swing has occurred, there is essentially static equilibrium in the lateral direction.

$\therefore \Sigma M_{\text{about the 5th wheel}} = 0$ where $\curvearrowleft$ +ve

$\therefore F_{Lateral} \cdot N - F_{centrifugal} \cdot M + M_{AJS} = 0$

$\therefore M_{AJS} = F_{centrifugal} \cdot M - F_{Lateral} \cdot N$

Examining $F_{centrifugual}$

$$F_c = \frac{m \cdot v^2}{r}$$

Where m = mass of the trailer
v = speed of the vehicle
r = radius of curve the vehicle is travelling on.

- As agreed upon at the steering committee, we would examine vehicles with trailer payloads ranging from 0 to 30,000 lbs (0 to 13,600 kg)

- The largest centrifugal forces will result from the heaviest trailer. For this reason, and the fact that it is the most common, we will examine only the 48 foot (14.6m) long trailer.

- As agreed to by the steering committee, we would examine vehicles travelling around curves at their design speed. Our investigations show that the design speeds seem to be based on a maximum lateral acceleration of 0.22 g. In this case, we will examine a 90 m radius, which has a design speed of 50 Km/h. Of course, vehicles may travel faster than the design speed, and as lateral acceleration is a function of speed squared, slight increases in speed may cause large increases in lateral acceleration. For this reason we will also examine these to some extent.

Empty Trailer Case (At design speed)

So, m = 6,260 Kg

$$v = 50 \text{ km/h} = 50 \frac{\text{km}}{\text{hr}} \times \frac{1 \text{ hr.}}{3600 \text{ sec}} \times \frac{1000 \text{ m}}{1 \text{ km}} = 13.9 \text{ m/sec}$$

r = 90 m

$$\therefore F_c = 6{,}260\text{kg} \cdot (13.9 \text{ m/sec})^2 \div 90 \text{ m} = 13{,}439 \text{ Newtons}$$

Loaded Trailer Case (At design speed)

m = 6,260 kg + 13,612 kg
= 19872 kg

v = 13.9 m/sec

r = 90 m

$$\therefore F_c = \frac{19{,}872 \cdot (13.9)^2}{90} = 42{,}660 \text{ Newtons}$$

Empty Trailer Case (at 0.3 g)

$$F_c = m \cdot a_{Lat} = 6260 \cdot 0.3(9.82)$$
$$= 18{,}442 \text{ Newtons}$$

Examining the Lateral Forces at the Trailer Tires

Rather than attempting to determine the lateral forces, the computer program was used to generate the lateral forces at each tire for various swing angles.

For the Empty Trailer Case

$$F_{Lateral} = F_7 + F_8 + F_9 + F_{10} = 96 + 88 + 126 + 115$$
$$= 425 \text{ Newtons}$$

For the Loaded Trailer Case

$$F_{Lateral} = 311 + 275 + 410 + 363 = 1,359 \text{ Newtons}$$

Substituting these values into the original Equation:

where $M_{AJS} = F_{centrifugal} \cdot M - F_{Lateral} \cdot N$

and

$M = 7.66$ m
$N = 12.3$ m

.·. For Empty Trailer Case (At design speed)

$$M_{AJS} = 13,439 \cdot 7.66 - 425 \cdot 12.3$$
$$= 97,715 \text{ N·m}$$

For Loaded Trailer Case (At design speed)

$$M_{AJS} = 42,660 \cdot 7.66 - 1,359 \cdot 12.3$$
$$= 310,000 \text{ N·m}$$

For Empty Trailer Case (At threshhold speed)

$$M_{AJS} = 18,442 \cdot 7.66 - 425 \cdot 12.3$$
$$= 136,038 \text{ N·m}$$

Examining the Resistive Moments required to Stop Trailer Swing at Different Swing Angles

As the previously calculated moments are quite large, it may be necessary to let the trailer swing somewhat so that the lateral forces increase. However, we must also contend with trailer rotational inertia if we do this.

Begin by examining the Moments about the Fifth Wheel.

$$\Sigma M = I\ddot{\Gamma}$$

$$\therefore F_{centrifugal} \cdot M - F_{Lateral} \cdot N - M_{AJS} = I\ddot{\Gamma}$$

Let's examine swing angles of 5°, 10°, 20°, 30°,

- for each angle there will be a new $F_{centrifugal}$ as velocity decreases and radius increases (assume negligible)
- $F_{Lateral}$ will change with the angle.

Determining the new $F_{centrifugal}$ for $\Gamma = 5°$

$F_C = \frac{m \cdot v^2}{r}$ v from computer simulation data @ $\Gamma = 5° = 13$ m/sec
$r \approx 90$ m

$\therefore F_C = 11{,}755$ N (empty trailer)

Determining the new $F_{Lateral}$ for $\Gamma = 5°$

From simulation data $F_{Lat} = 539 + 425 + 539 + 425$
$= 1{,}928$ N

Determining the Moment of Inertia (Empty Trailer)

$I_{C \text{ of } G} = 150{,}194 \text{ kg} \cdot \text{m}^2$

From Parallel Axis Theorem $I_{5th\ wheel} = I_{CofG} + m \cdot d^2$

where d = distance from center of mass to 5th wheel $= I_{CofG} + m \cdot M^2$

$= 150{,}194 + 6260 \cdot (7.66)^2$

$\therefore I = 517{,}503 \text{ kg} \cdot \text{m}^2$

Determining Decceleration Rate

Let's assume that the AJS engages about halfway through the angle we intend to let it swing over. I will say that the articulation rate at this point is the maximum (or $\dot{\Gamma}_i$).

Now Torque = $I\ddot{\Gamma} = I\frac{(\dot{\Gamma}_f - \dot{\Gamma}_i)}{\Delta t}$ ← Constant Deceleration

- I assume there will be constant deceleration (not quite true).
- I assume $\dot{\Gamma}_i \approx$ articulation rate when the trailer has reached the halfway point through its swing angle.

$\therefore \dot{\Gamma}_{2.5°} \approx 6.4°/sec = 0.112$ rad/sec

- I assume that the time to reach the desired swing angle will be the remaining angle divided by the average rotational speed ($\dot{\Gamma}_i$ / 2)

$\therefore \dot{\Gamma}_i$ / 2 = 6.4°/sec ÷ 2 = 3.2°/sec = .056 rads/sec

and

$$t = \frac{2.5°}{3.2°/sec} = 0.78 \text{ secs}$$

$$\therefore I\ddot{\Gamma} = 517{,}503 \text{ Kg}\cdot\text{m}^2 \cdot \frac{(0 - 0.112 \text{ rad/sec})}{0.78 \text{ secs}}$$

$$= -74{,}308 \frac{\text{Kgm}^2}{\text{s}^2}$$

Substituting into the original Equation

$$F_{cent} \cdot M - F_{Lat} \cdot N - M_{AJS} = I\ddot{\Gamma}$$

$$M_{AJS} = 11{,}755 \cdot 7.66 - 1{,}928 \cdot 12.3 - (-74{,}308)$$

$$= 140{,}637 \text{ N}\cdot\text{m}$$

Repeating this method for Γ = 10°, 20°, and 30° resulted in:

$\Gamma = 10°$ $M_{AJS} = 117{,}000$ N·m

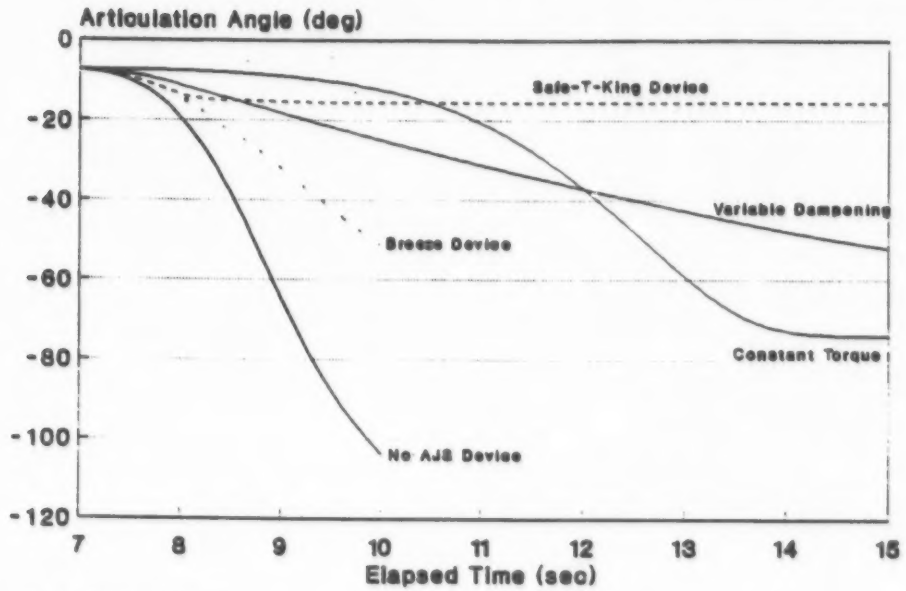
$\Gamma = 20°$ $M_{AJS} = 48{,}000$ N·m

$\Gamma = 30°$ $M_{AJS} = 26{,}000$ N·m

As the method used to derive these values is quite crude, these values are approximate only.

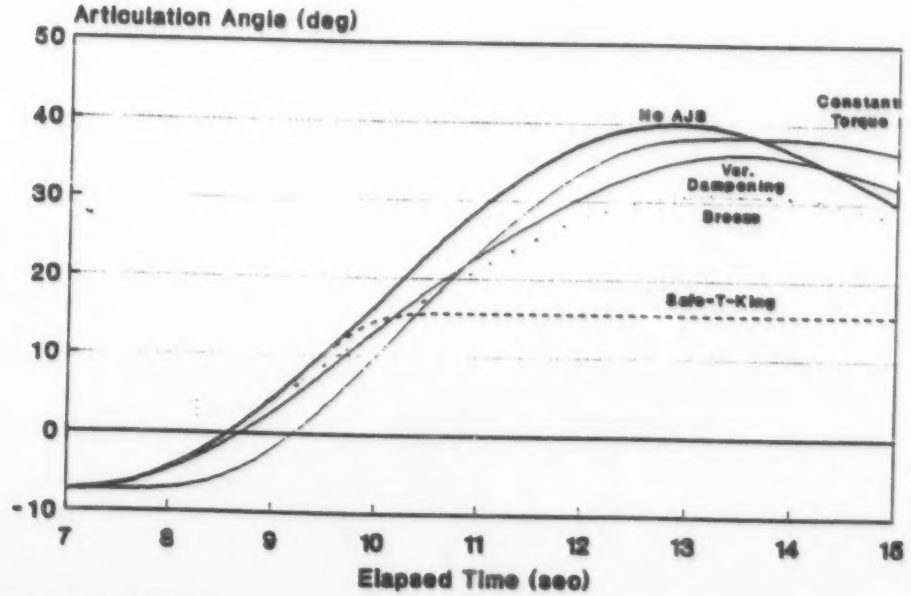
APPENDIX "G"
AJS Plots (expanded time frame)

ARTICULATION ANGLE VERSUS TIME
(Tractor Jackknife)



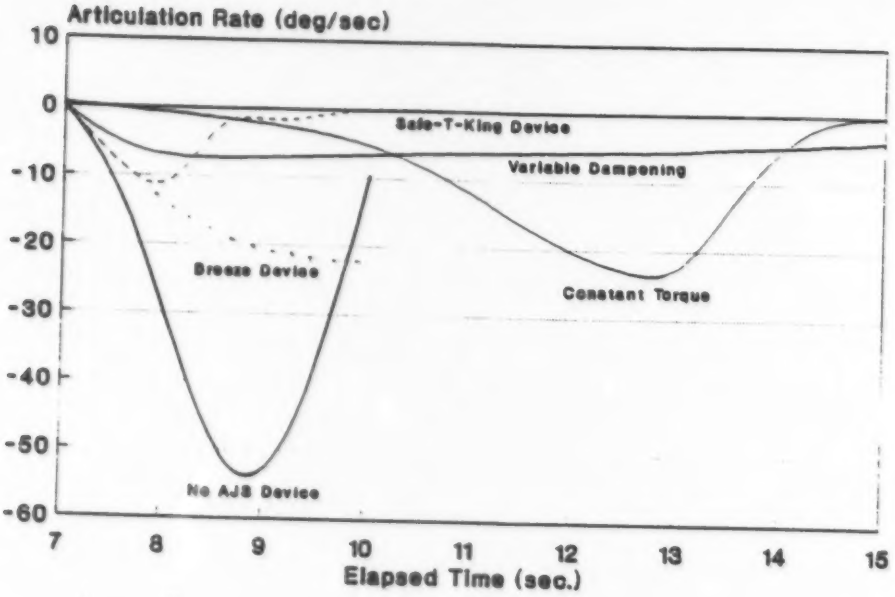
Constant Radius Turn

ARTICULATION ANGLE VERSUS TIME
(Trailer Swing)

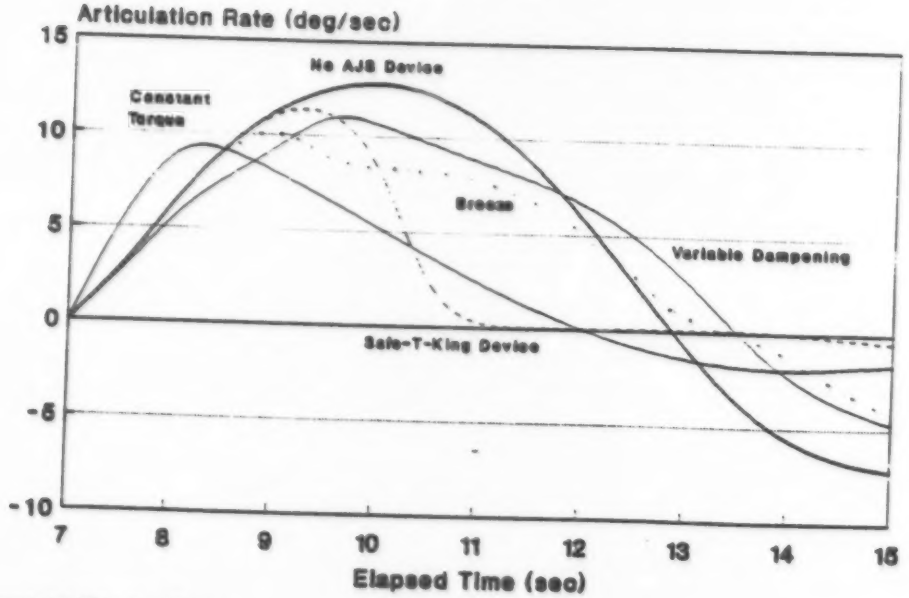


Constant Radius Turn

ARTICULATION RATE VERSUS TIME
(Tractor Jackknife)

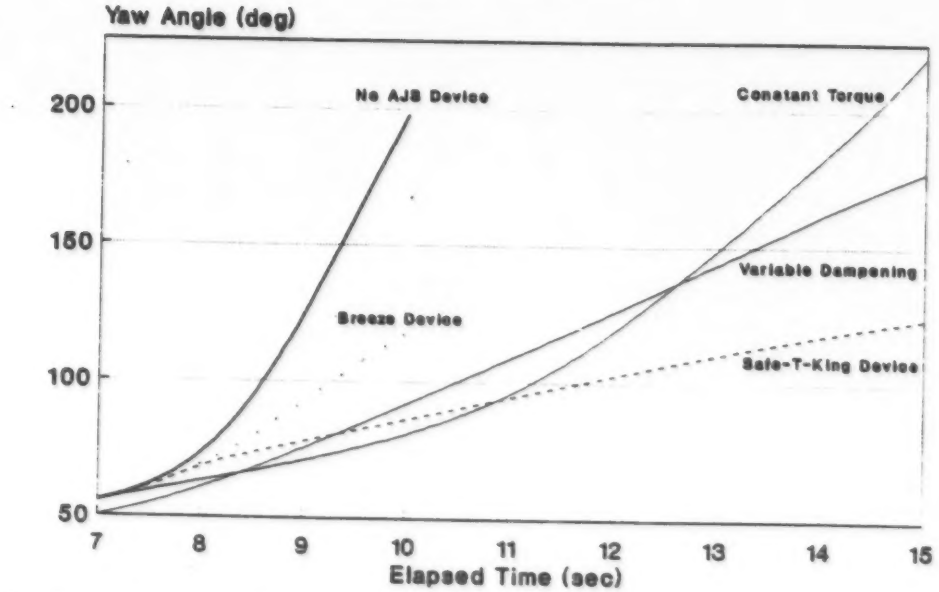


ARTICULATION RATE VERSUS TIME
(Trailer Swing)



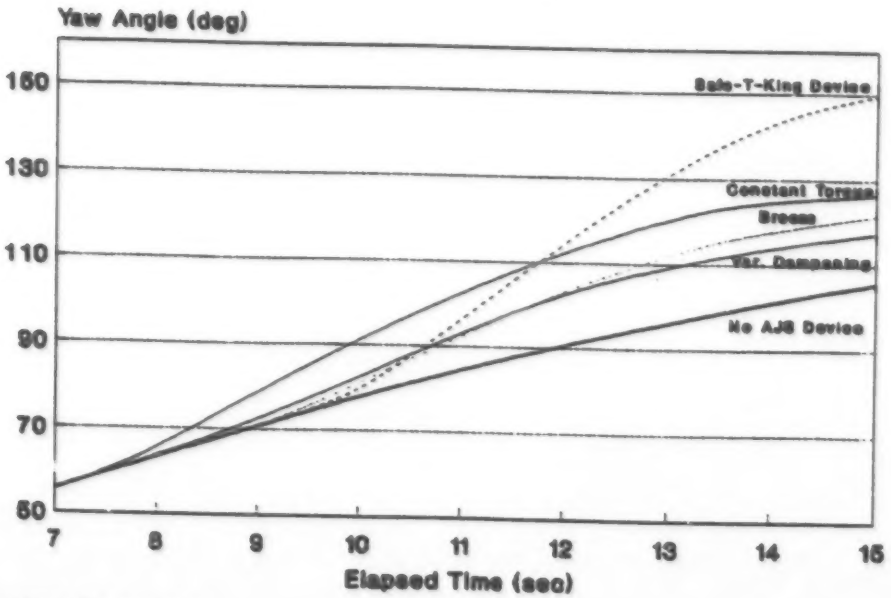
Constant Radius Turn

YAW ANGLE VERSUS TIME
(Tractor Jackknife)



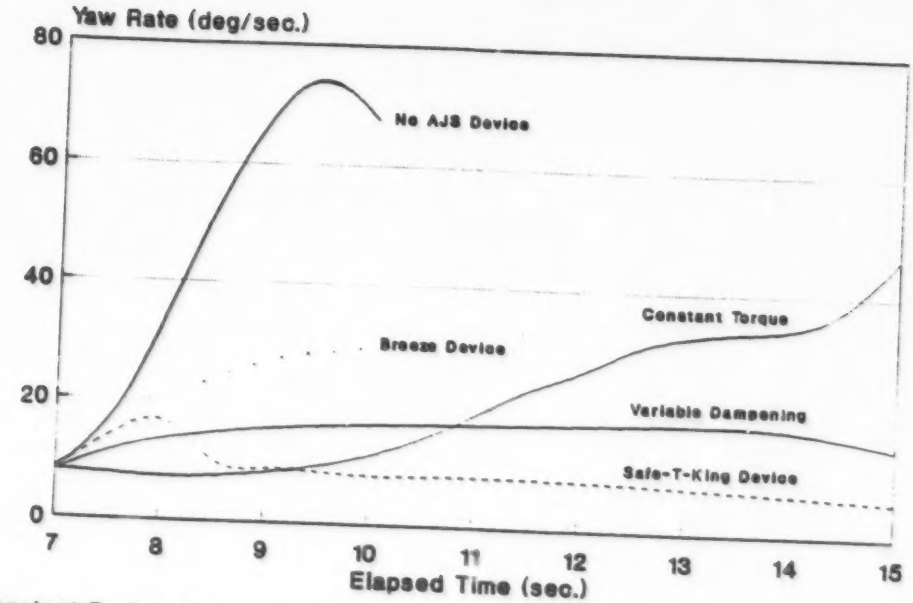
Constant Radius Turn

YAW ANGLE VERSUS TIME
(Trailer Swing)



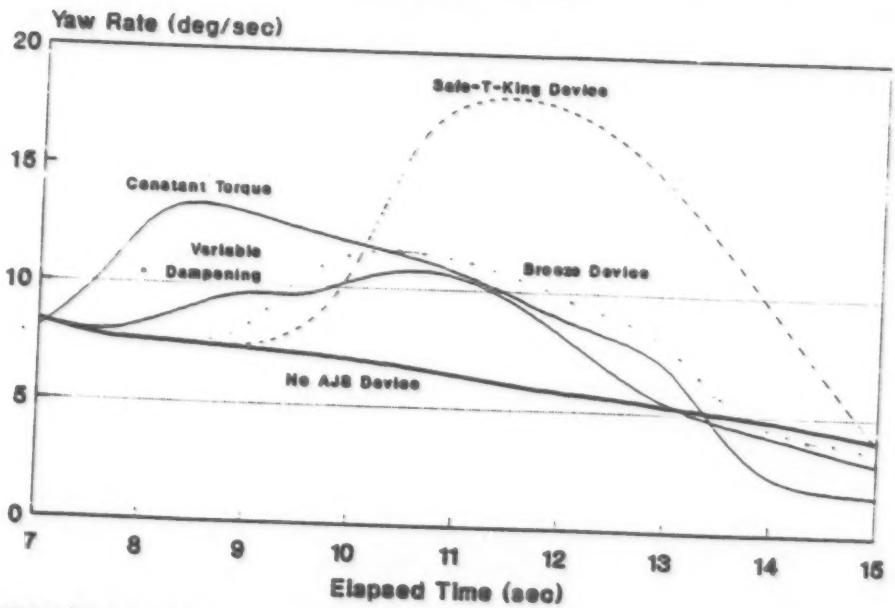
Constant Radius Turn

YAW RATE VERSUS TIME
(Tractor Jackknife)



Constant Radius Turn

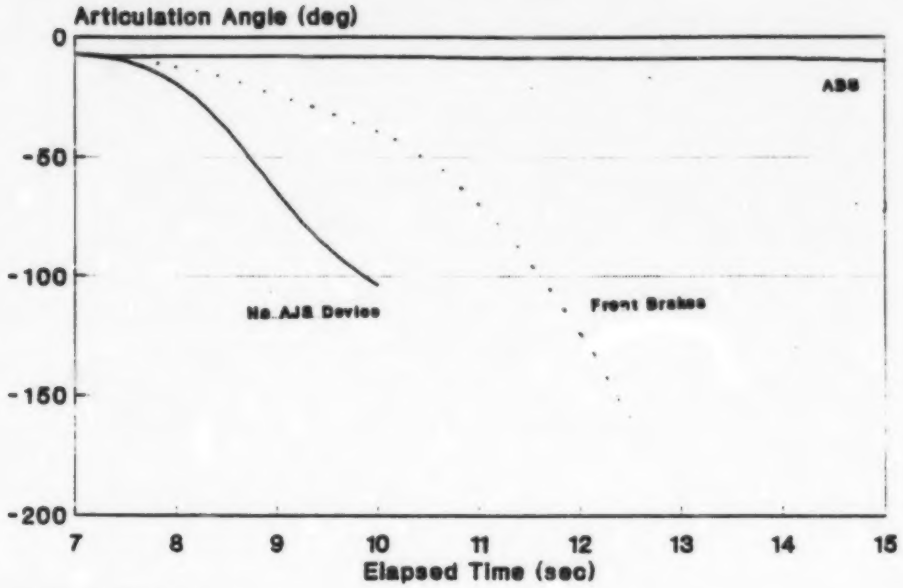
YAW RATE VERSUS TIME
(Trailer Swing)



Constant Radius Turn

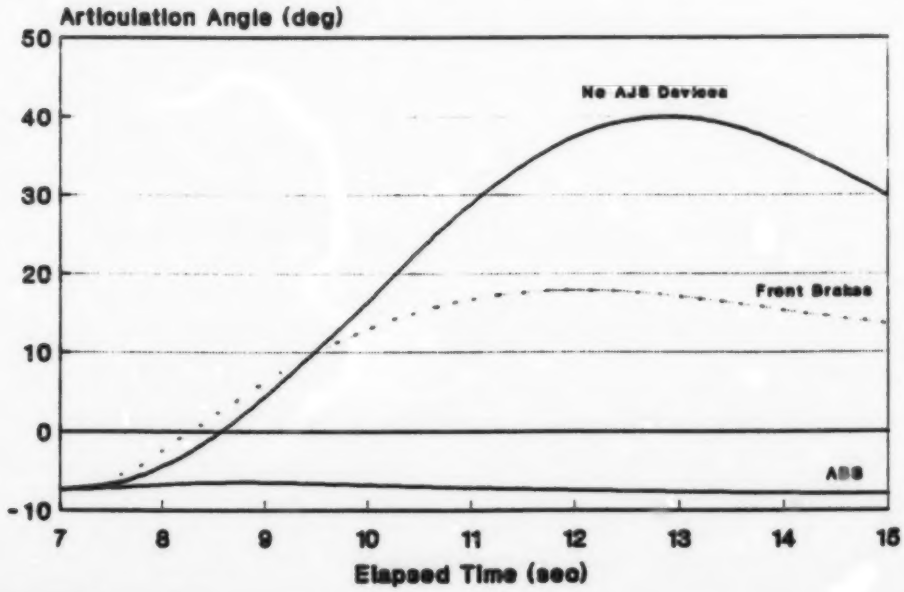
APPENDIX "H"
ABS and Front Brake Plots

ARTICULATION ANGLE VERSUS TIME
(Tractor Jackknife)



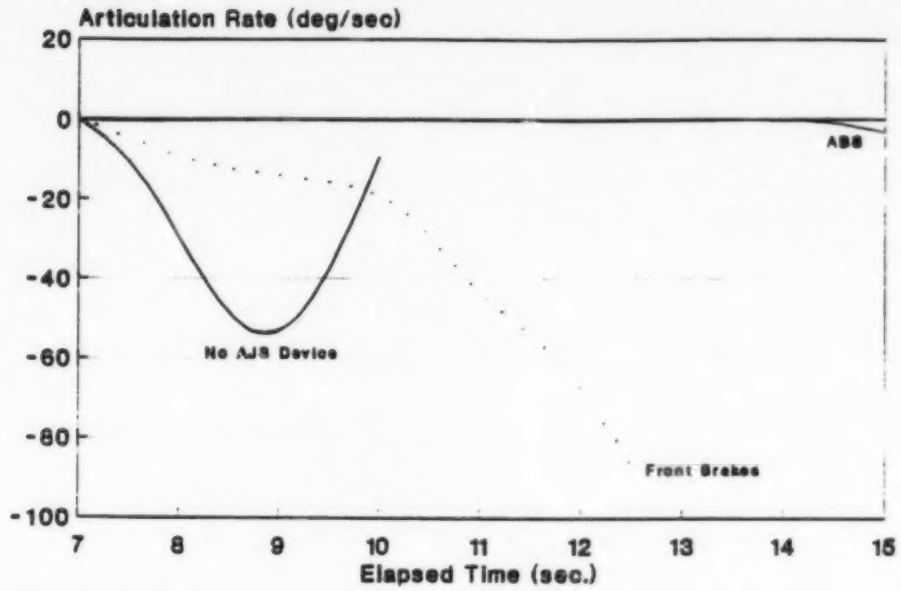
Constant Radius Turn

ARTICULATION ANGLE VERSUS TIME
(Trailer Swing)



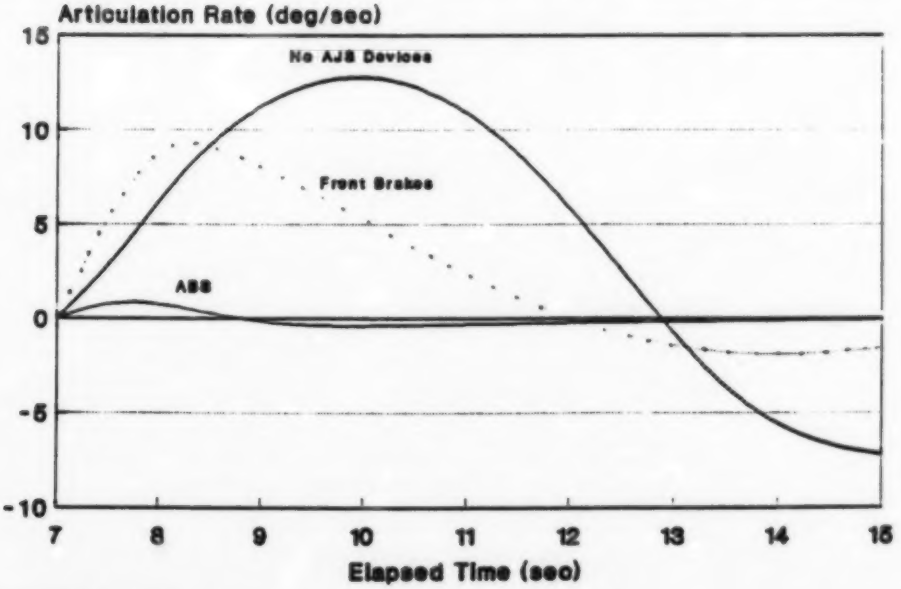
Constant Radius Turn

ARTICULATION RATE VERSUS TIME
(Tractor Jackknife)



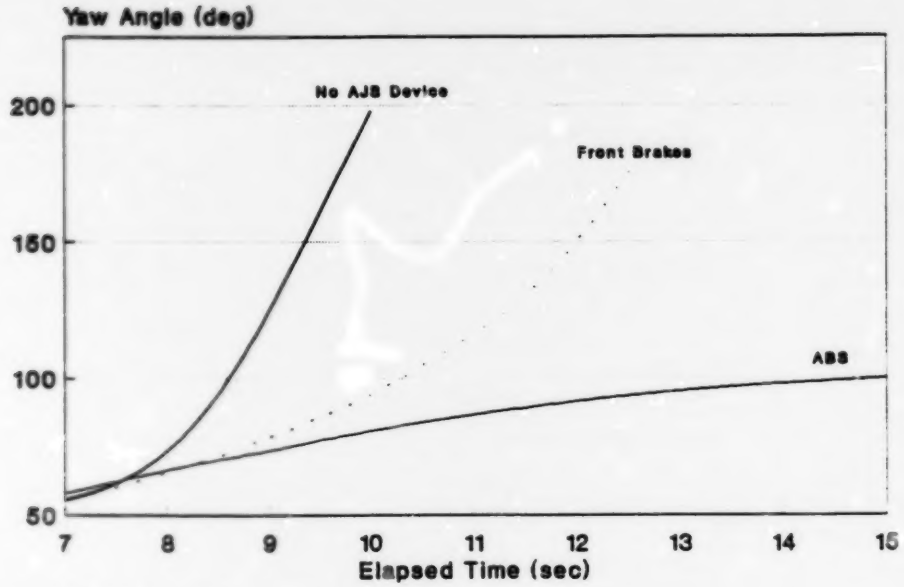
Constant Radius Turn

ARTICULATION RATE VERSUS TIME
(Trailer Swing)



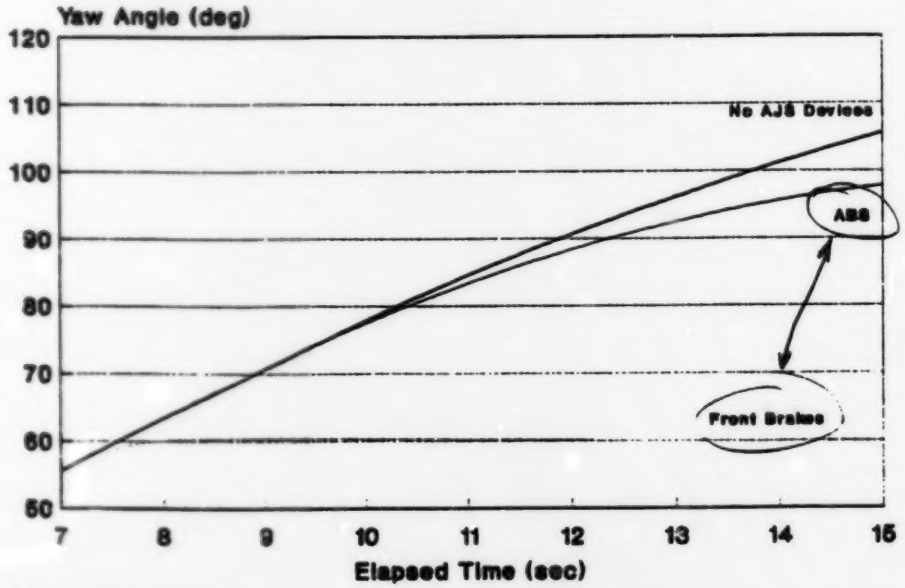
Constant Radius Turn

YAW ANGLE VERSUS TIME
(Tractor Jackknife)



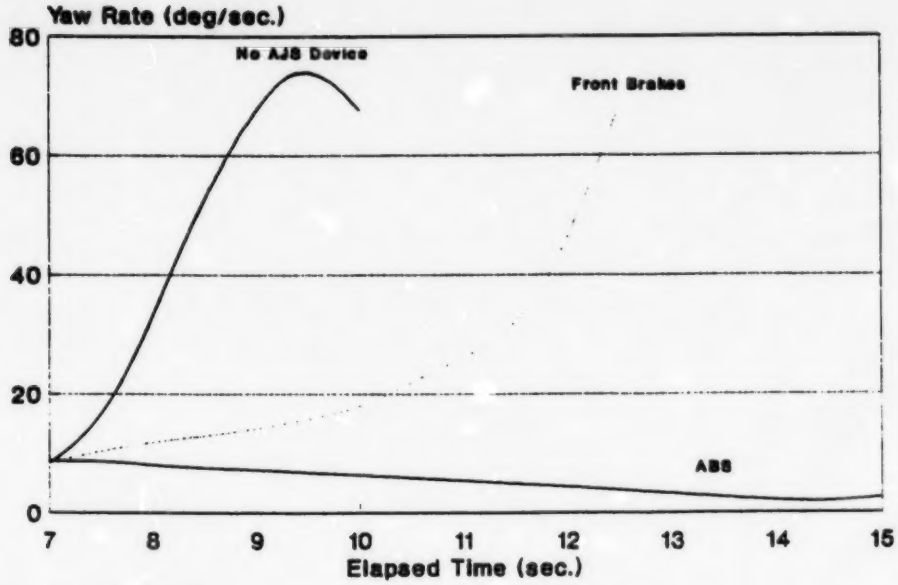
Constant Radius Turn

YAW ANGLE VERSUS TIME
(Trailer Swing)



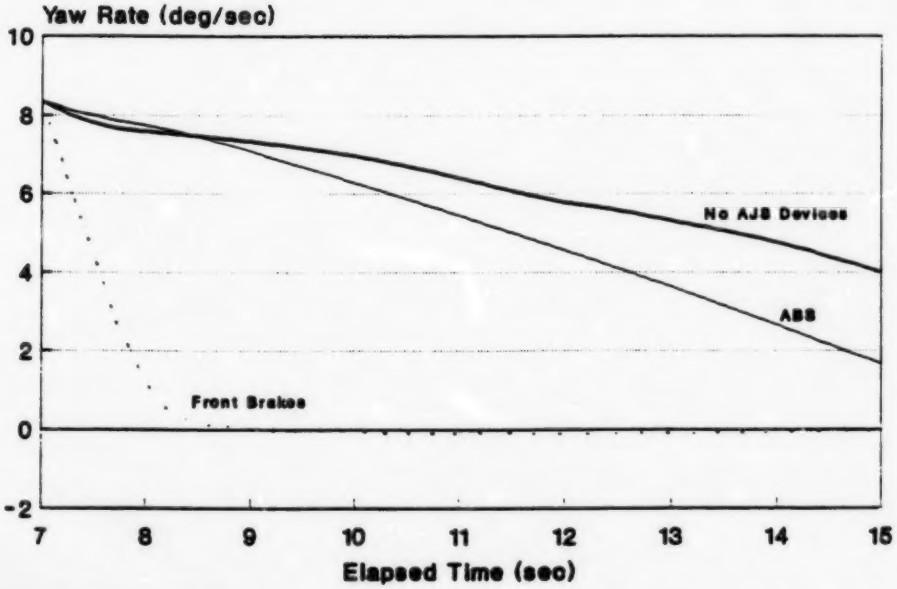
Constant Radius Turn

YAW RATE VERSUS TIME
(Tractor Jackknife)



Constant Radius Turn

YAW RATE VERSUS TIME
(Trailer Swing)



Constant Radius Turn